IND

HOT ROD

MAGAZINE YEARBOOK

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PETERSEN PUBLISHING COMPANY

No. 1

By The Greatest Names In Hot Rodding—

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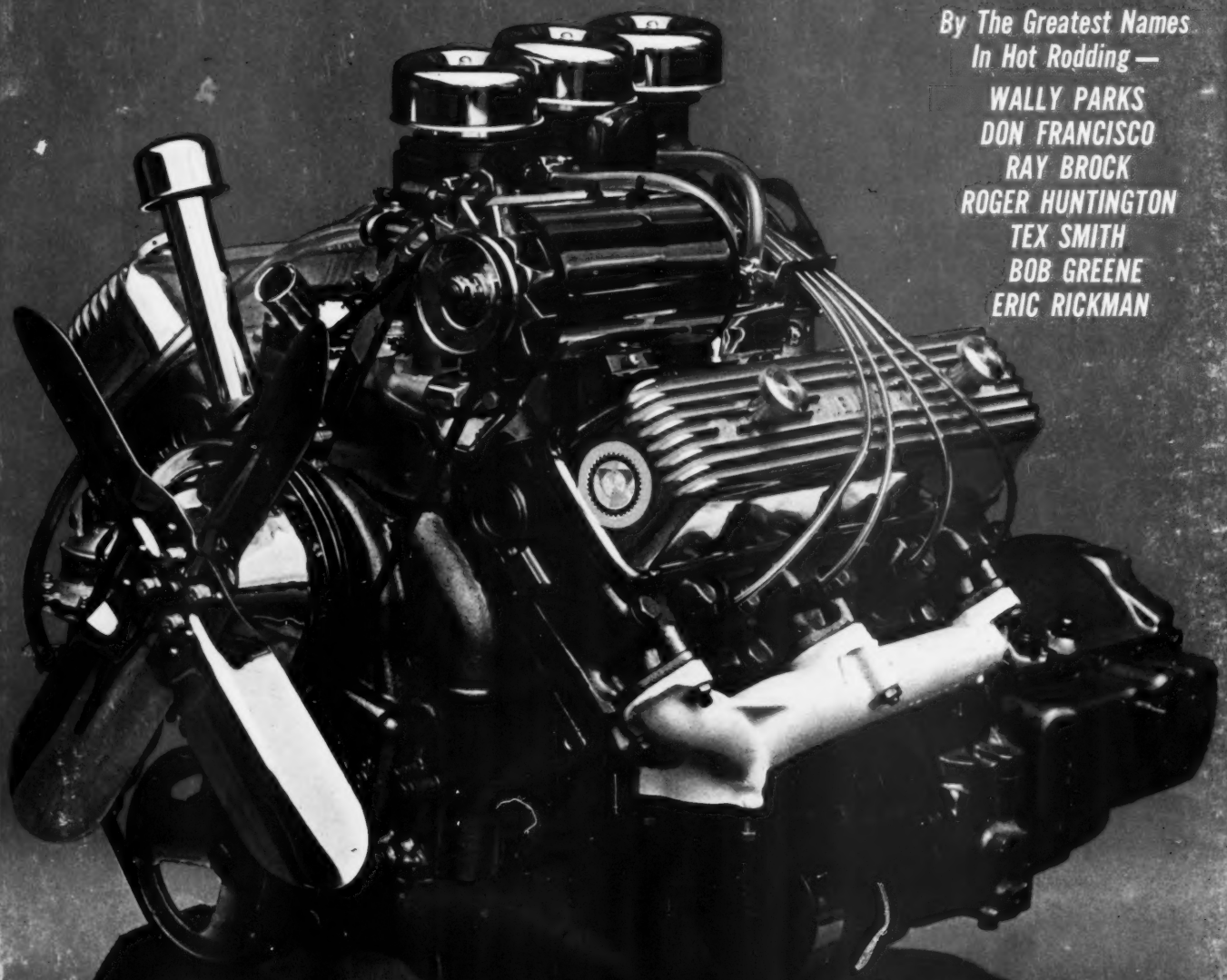

SUPER STOCK ENGINES • HOT IGNITION • COMPACT CAR RODDING
TRANSMISSION TIPS • DRAGS, RAILS, ROADSTERS • CARBURETION

FULL YEAR NHRA RECORDS AND PHOTOS

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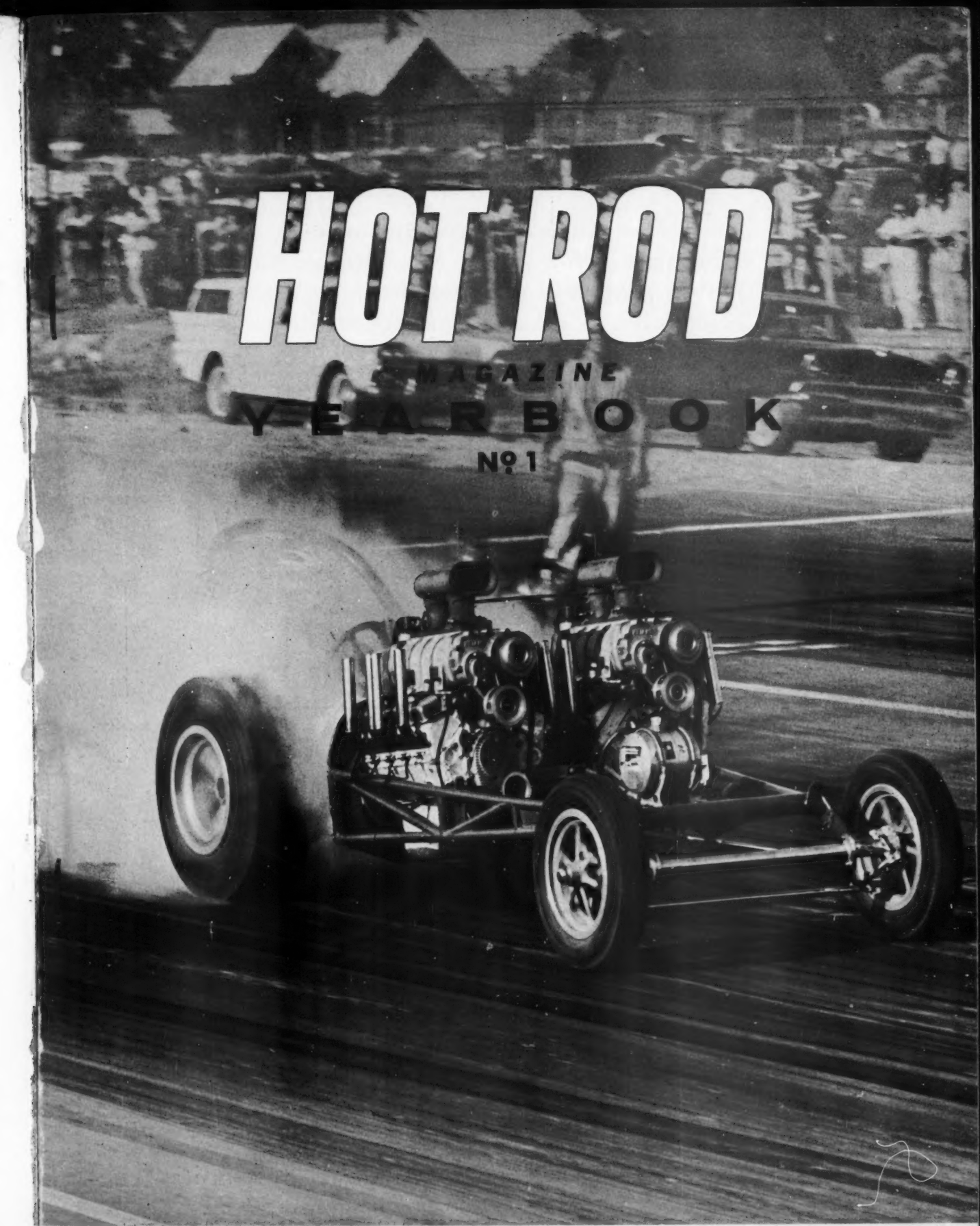

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Nº 1

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Cover photo of full house Olds street engine by Eric Rickman

HOT RODDING 'JUST HAPPENED'

Nobody invented hot rodding. It just happened along, as one of the automobile's inherent "available options." Ever since the days of the early Model T, and even before then, automotive pioneers were busy in backyard sheds, creating new kinds of horseless carriages and using "imagination" as their most effective blueprint.

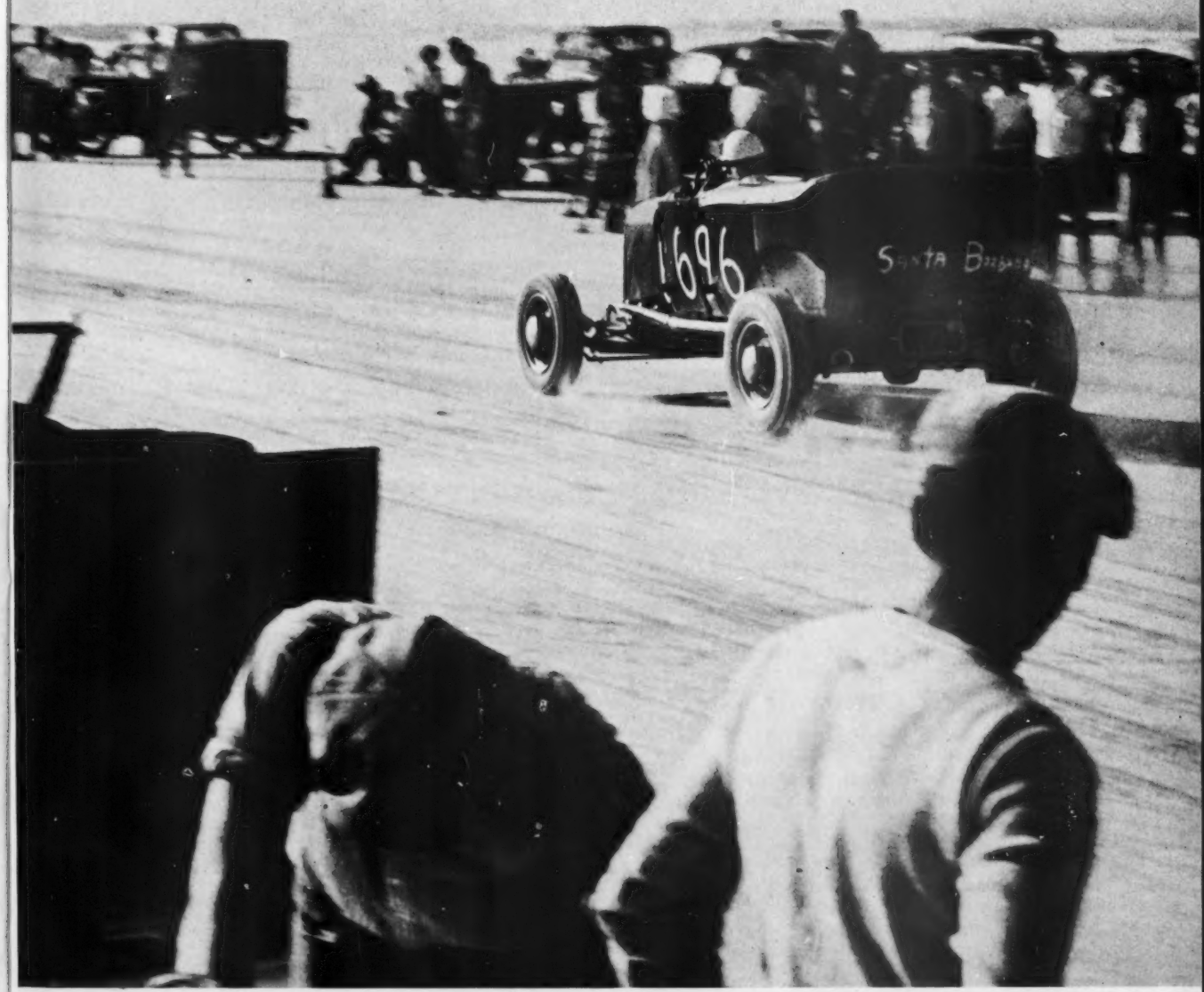
Today's hot rodder is much like his tinkerer ancestor, except that his challenge to produce something really "different" is a bit more complicated. In a mechanical world where auto manufacturers wage endless battles to stay ahead in engineering and styling, the hot rodder is hard pressed in his desire to create anything truly original.

Many of the components and features found in today's passenger cars can be traced back to hot rodders. Although the "ideas" were not new to the industry by any means, it was the persistent application and usage by the hot rodder that brought many an "impractical" idea into popularity and into production car lines.

As someone once remarked in an S.A.E. meeting at Indianapolis in 1952, "The hod rodder doesn't realize all these things are 'impossible' so he does them!"

This Yearbook is dedicated to the hot rodder—yesterday's, today's and tomorrow's. Whether his interest lies in building from scratch, re-creating an old model, restyling a later model, developing a powerful charger capable of setting new performance records, or just plain dreaming and wishing, this is his book.

WALLY PARKS



THE NEW SUPER STOCK ENGINES

DURING the first few months of the 1961 automobile model year several automobile manufacturers announced the availability of optional high-performance engines and parts for their cars. The reasons the individual factories had for producing this equipment aren't important to a hot rodder but the limited production and consequent scarcity of some of it, especially complete engines, would lead one to believe that the primary reason was to provide professional stock car racing crews with equipment that would uphold the factory's name against competing makes.

It seems that the results of stock car races have a measurable influence on many purchasers of new automobiles. Many men like to drive and be associated with the make of car that won at Daytona, Darlington, or at some other major stock car track although they may never drive their cars over the legal speed limit. In other words, most high-performance optional engines and parts are produced by the factories more for advertising purposes than for public consumption. However, it doesn't make any difference to a hot rodder whether an engine or a part was built to enhance the value of a car in the eyes of the public, although the public may never own or even see the part, or whether it was built especially for him if he can get his hands on the parts he needs. Most drag racing associations have accepted the optional engines as being eligible to run in Super Stock classes.

Actually, there's nothing new about optional factory high-performance engines. They have been available from Chevy, Ford, Pontiac, Dodge, Plymouth, etc., for several years. But, naturally, the 1961 versions are the biggest and strongest.

CHEVROLET

Chevrolet has a long list of optional engines for 1961 but the three most important ones are the 340, 350, and 360 horsepower models.

The 340 hp engine has the same bore and stroke as other 348's but it has different pistons that give it a compression ratio of 11.25 to 1. Its cylinder heads differ from all the other lower horsepower versions except one rated at 305 hp by having larger intake and exhaust passages and valves. Although the passages are larger than those in other heads the ports at the ends of the passages are the same sizes for all heads. The intake valves are ⅛-inch larger than those in standard heads and the exhausts are 1/16-inch larger. The larger passages and valves increase the engine's breathing ability considerably. These heads can be used on any 348 cylinder block without being changed or altered in any way.

The engine's camshaft is a solid-tappet type ground to provide much more radical valve action than the hydraulic lifter shafts that are standard in the less powerful engines. Dual valve springs stronger than those used with hydraulic-lifter shafts make the valves and their actuating mechanism follow the cam at the higher crankshaft speeds the camshaft makes possible.

Carburetion is provided by a single four-throat carburetor on an aluminum manifold. This carburetor differs from the standard four-throat by having larger venturis and throttle bores. The manifold differs from the standard cast-iron manifold by having larger risers to match the bores in the carburetor.

As the engine is not designed for street driving, its ignition distributor does not have a vacuum advance diaphragm. The purpose of a vacuum diaphragm is to increase an engine's total spark advance at low and medium crankshaft speeds when the engine is running under light-load conditions. The high vacuum in the engine's intake manifold under these conditions actuates the diaphragm to advance the timing. A high-performance engine is not designed to run for long periods of time under light-load conditions; therefore, its distributor does not need a vacuum advance diaphragm.

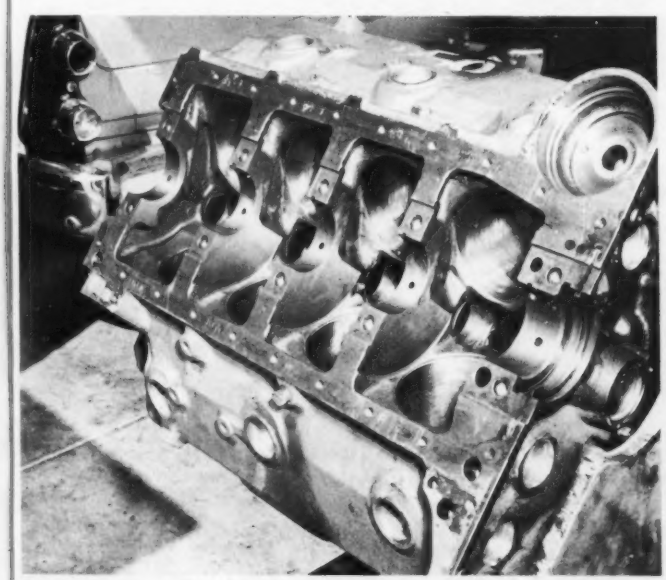
Although the ignition system uses a single coil, its distributor is fitted with dual-points. The points are arranged so their closed periods overlap. This increases the degrees of dwell, which is the period of time the points are closed so that primary current can flow through the ignition coil, for each of the engine's cylinders while at the same time allowing each set of points to be opened far enough to minimize arcing across the gap between their contacts. The ignition system also has a coil that differs from the standard coil.

The 350 horsepower version of the 348 is exactly the same as the 340 except that it has three two-throat carburetors on a special aluminum manifold instead of the single four-throat. This three-carburetor setup is altogether different from the one used on lower-rated engines. The venturis in the standard setup's carburetors are smaller and its manifold is cast-iron.

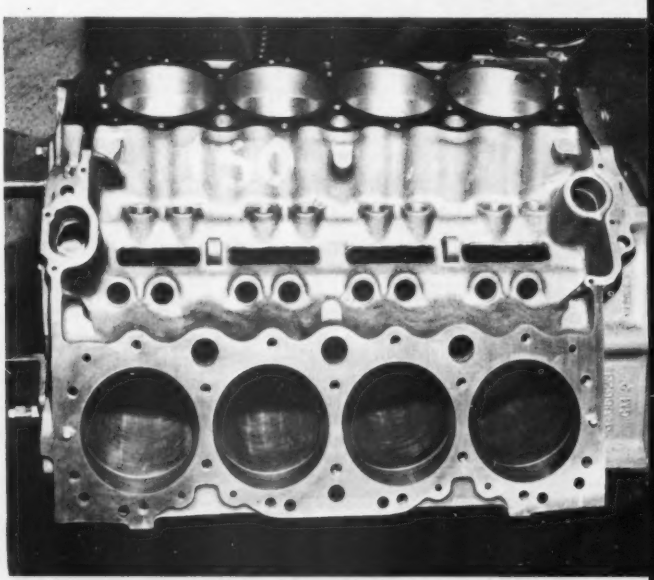
Chevy went all-out on its 360 horsepower '61 option. This is the new 409 engine that has been grabbing the Super Stock trophies at the many drag strips where it has run; however, one or two 390 Ford's are holding their own against the 409's now.

The 409 gets its model designation from its displacement. With the exception of modifications made to it to accommodate the larger displacement, the 409 cylinder block is identical to 348 blocks. Coring for it was changed to enable its cylinders to be enlarged from the 348 diameter of 4⅛ inches to 4 5/16 inches without sacrificing cylinder wall strength. The only other change to the block is in its crankcase. Sides of its main bearing webs were machined to provide clearance for the thicker, heavier counterweights on the 409's crankshaft.

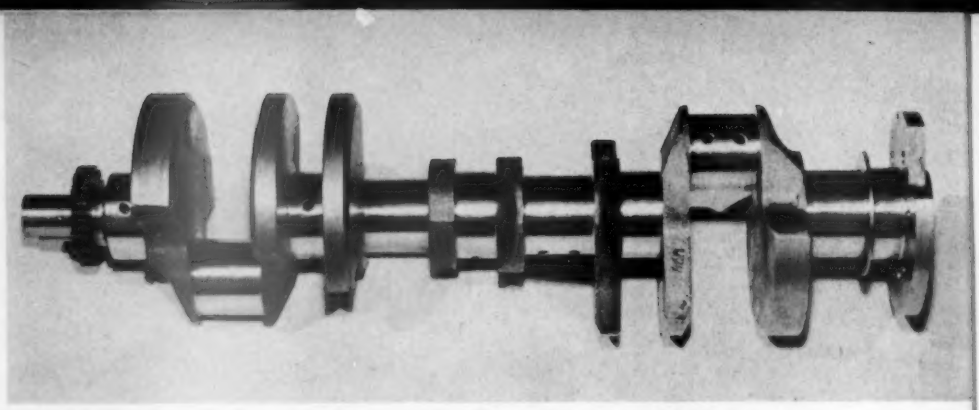
Newest of the Super Stocks for 1961 is Chevy's 409. Engine is similar to the 348 except for its larger bore and longer stroke.



Sides of the main bearing webs in 409 blocks are machined to provide clearance for the crankshaft's thicker counterweights.

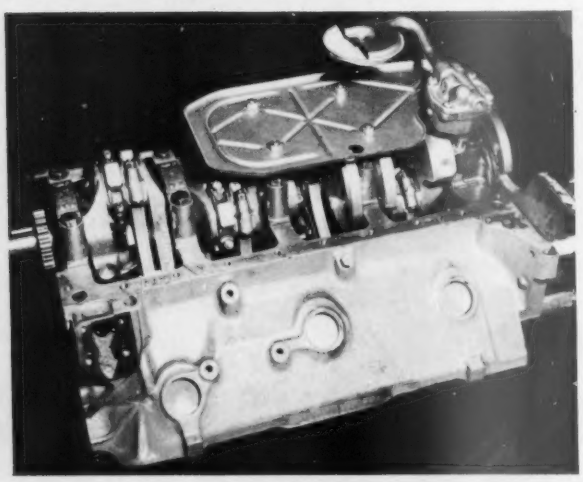


Combustion chambers in 409 are same as 348's except they don't have cutouts for valve clearance, compression ratio.



Main bearing journals and crankpins on the crankshaft for the 409 have the same diameters as those on 348 shafts but the 409's counterweights are thicker and heavier.

Pistons in the 409 are forged aluminum for maximum strength. They have small squish areas, full-length valve relief areas.



Oil in the 409's crankcase is kept at a lower temperature and under control with this baffle and another attached to the pan.

It was necessary to make the counterweights on the crankshaft heavier to compensate for the shaft's 3½-inch stroke. Stroke length in 348's is 3¼ inches. Weight of the 409 shaft is 67 pounds, compared to the 59 pounds of a 348 shaft. Main bearing journals and crankpins on 409 shafts have the same diameters and lengths as those on 348 shafts. Something the 409 shaft has that the 348 doesn't is an additional counterweight on its flywheel flange. Oldsmobile, Buick, and Pontiac have been using additional counterweights of this type for some time now.

Pistons in the 409 follow standard racing practice by being forged aluminum rather than cast aluminum. Chevrolet prefers the term "impact extruded" rather than forged but forged is a more familiar term to hot rodders. Heads of the pistons have a full-width machined area to provide valve head clearance, and a very narrow squish area. The heads are shaped to reduce the combustion chamber volume to that required for an 11.00 to 1 compression ratio.

Pistons for the 409 also differ from those for 348's by having their piston pin bores midway between their skirts. The bores in 348 pistons are offset .060-inch toward the major thrust side to allow the pistons to operate more quietly but in a racing engine piston noise loses its importance.

Connecting rods for 409's have a ⅛-inch shorter center to center length than 348 rods to compensate for the crankshaft's ¼-inch longer stroke. But although the rods are shorter, they are heavier than 348 rods. This additional weight is a result of adding material to their ends and their shank to increase their strength. Piston pins are a press-fit in them, as they are in all Chevy V8's.

Cylinder heads for the 409 are machined from the same castings used for 340 and 350 horsepower 348's. In fact, with the exception of different spot-facing around their valve guides and larger pushrod openings, 409 heads are identical to those for 340 and 350 hp engines. The different spot-facing was necessary to accommodate the single valve springs that are used in place of the 348 dual springs. These single springs are stiffer than the duals. Larger pushrod openings were necessary for ⅜-inch diameter rods used in place of the 5/16-inch rods that are standard in 348's. The purpose of the larger diameter rods was to better withstand the greater pressure of the 409 valve springs.

Valves in the 409 are actuated by a camshaft that has the most radical action of any of the factory shafts for the big Chevy's. It is, of course, designed for solid valve lifters. It is driven by a standard 348 sprocket and chain assembly.

The aluminum intake manifold for the 340 hp 348 was adapted to the 409 by enlarging its risers for the primary throats in a larger carburetor. The carburetor is a Carter 3270S that has a total venturi area of 6.67 square inches and a 5-inch diameter air inlet. Fuel is supplied to it by a pump that has a pressure range of 9.25 to 10.75 pounds. This pump is also used on 340 and 350 hp 348's.

The same ignition distributor that is standard on 340 and 350 hp 348's is also used on the 409. The 409 also has a special ignition coil.

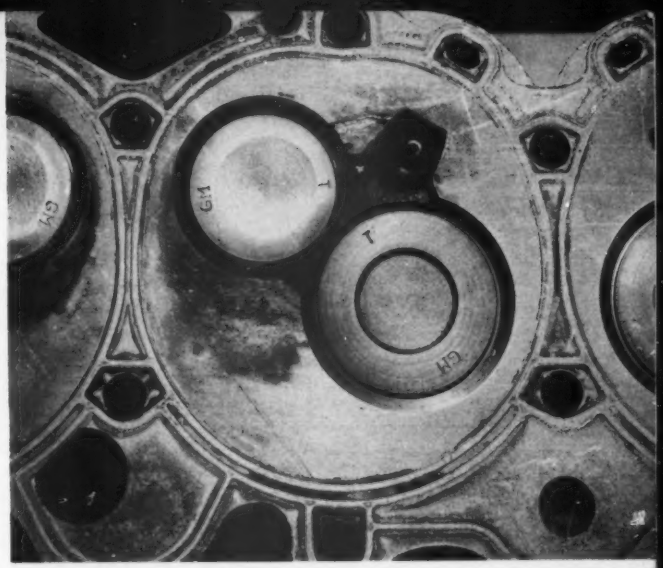
To keep the 409's lubricating oil cooler and under better control when the engine is used in track racing cars the capacity of its oil pan was increased from the four-quart 348 capacity to six quarts and the crankcase was baffled. The baffling consists of two sections, one of which is attached to the pan and the other to extensions on four of the cap-screws that secure two of the main bearing caps to the cylinder block. The pickup tube on the oil pump is fixed rather than of the floating type. This helps keep it below the surface of the oil in the pan's sump at all times. A full-flow oil filter is standard equipment.

All of Chevy's high-performance V8's prior to the 409 have been fitted with Moraine 400 main and connecting rod bearing inserts. Inserts in the 409 are Moraine 500, which are described as an improved version of the 400 inserts. Both types of bearings consist of a steel shell that supports an aluminum matrix. This matrix is porous and impregnated with a lead alloy bearing material. Durability of the 400 inserts is excellent; the 500's must be better in some way than the 400's but they will be entirely satisfactory if their durability is even equal to that of the 400's.

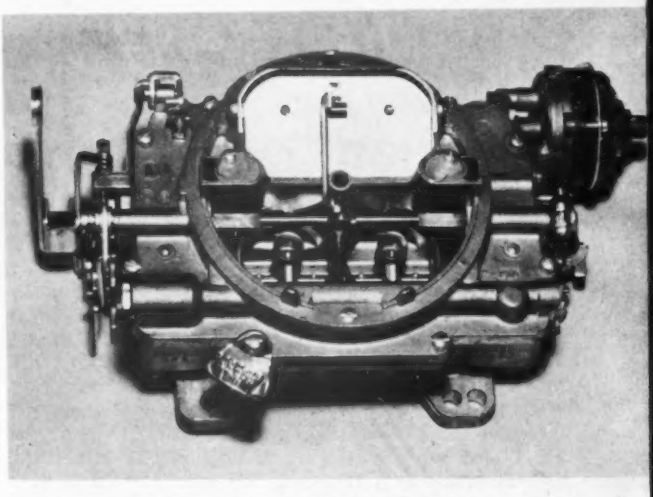
CHRYSLER

Each Chrysler Corporation car has its own optional high-performance engine but these became more or less obsolete early in 1961 when the Corporation announced that their 413 cubic inch Golden Lion V8 was available on special order in most of their full-size passenger cars. With the 413 available as optional equipment there wasn't much sense in buying a hotted-up version of a smaller engine. The 413 is rated at 350 hp at 4600 rpm. Its torque rating is 470 pounds-feet at 2800 rpm.

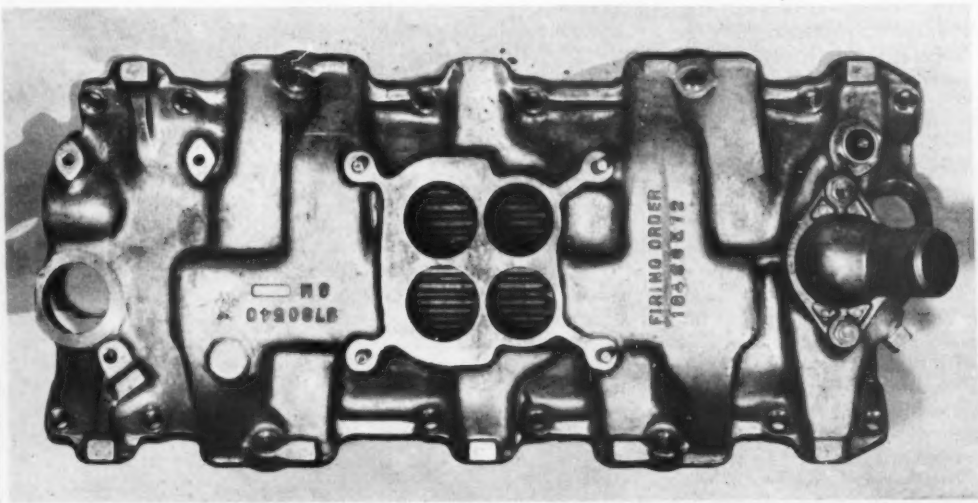
Although the 413's displacement is exceeded only by the 430 cubic inches of Lincoln's engine it is not a true high-performance power plant. It is strictly a passenger car engine built with enough displacement to pull the Corporation's large and heavy Chrysler and Imperial automobiles. It does not have any special features, either internally or



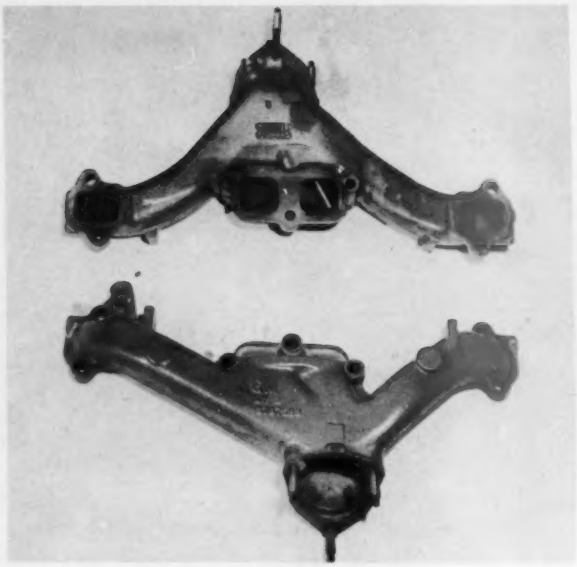
Although combustion chamber surfaces of 348 and 409 cylinder heads are said to be flat, they have recesses for the valves.



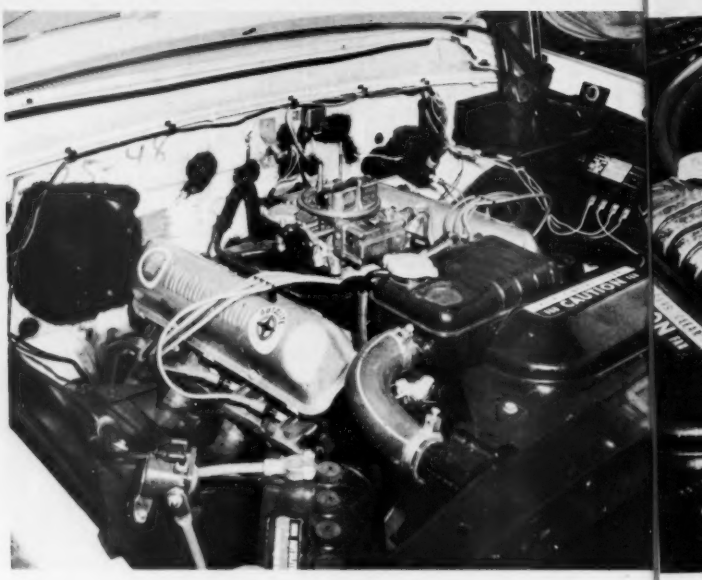
The Carter four-throat carburetor that is standard equipment on the 409 has more venturi area than any 348 4-throat carb.



An aluminum intake manifold for a 350 hp 348 engine was adapted to the 409 by enlarging two of its carburetor riser throats.



Exhaust manifolds for the 409 are fairly well streamlined for easy gas flow but a good set of headers could be more efficient.



Ford's first 390 cubic inch high-performance engines were fitted with a single four-throat carburetor, were rated at 375 hp.

externally, to allow it to run at the high crankshaft speeds necessary for high power output; however, it cannot help but have good torque and power outputs because of its large displacement. The thought trying to be conveyed here is that the engine performs well but in its stock form its torque and power output potentials are untapped. If Chrysler chose to give its internal parts racing clearances, possibly make some of them stronger for higher crankshaft speeds, install a solid-lifter camshaft with the necessary timing and lift, and add some carburetion, this engine, in some of the Corporation's smaller cars, could become a real menace to the competition.

Chrysler builds a semi high-performance 413 engine for their 300 series cars but this is nothing more than a standard 413 with a different camshaft and two four-barrel carburetors. The camshaft lifts the valves a little higher and holds them open a little longer but it uses hydraulic valve lifters. There isn't anything wrong with hydraulic lifters for passenger car engines but their inability to function at high cycling speeds makes them unsuitable for high-performance power plants. The 300G engine for '61 is rated at 375 hp at 5000 rpm. Its torque output is said to be 495 pounds-feet at 2800 rpm.

FORD

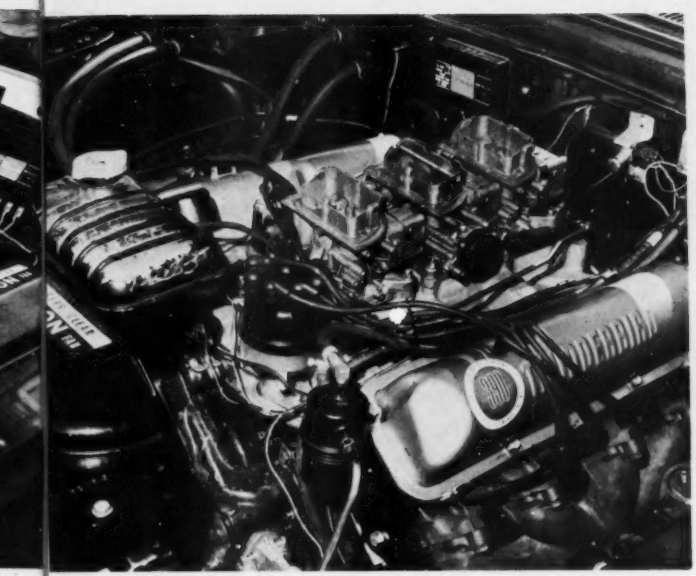
Ford has been a steady contender in the high-performance race. They started 1961 with a good 390 cubic inch engine that could easily have been championship material at the drag races but before the fellows really got them going Chevy blew the works with their new 409. In the Los Angeles area the 409's, after they first came out, led the Ford's awhile but now one or two Ford's are collecting their share of wins. Pontiac's are pushing both the Ford's and the 409's but usually they settle for third. On the longer stock car tracks where the cars can let it all hang out the Pontiac's have been leading the Chevy's and Ford's, with the Chevy's running behind the Pontiac's but well ahead of the Ford's.

Ford has two 390 cubic inch engines for 1961. One is for standard production passenger cars and the other is a high-performance option. In exterior appearance the engines are similar but inside they differ considerably. The standard production model is rated at 300 horsepower at 4600 rpm, with 427 pounds-feet of torque at 2800 rpm, and the high-performance option is rated at 375 horsepower at 6000 rpm with 427 pounds-feet of torque at 3400 rpm.

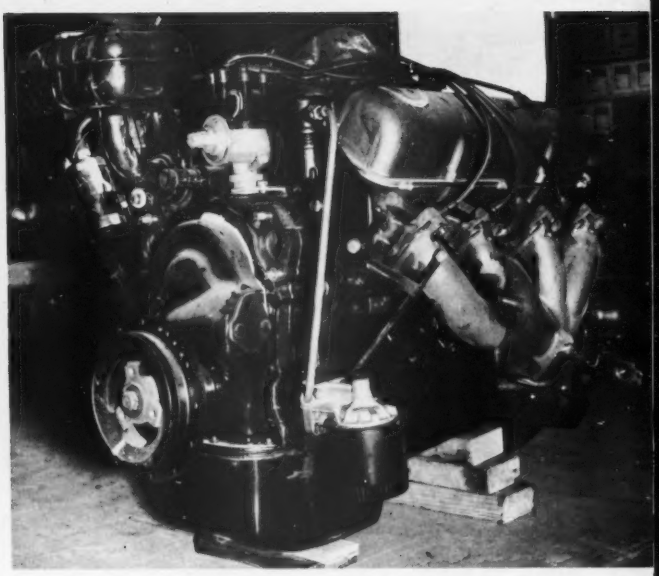
Both engines have the same 4.05-inch bore and 3⅞-inch stroke but the 300 hp engine has a 9.60 to 1 compression ratio in comparison to the 10.60 ratio of the 375 hp high-performance option. The 375 has a solid-tappet camshaft that provides more radical valve action than the hydraulic-lifter shaft in the standard engine. Both engines have a single four-throat carburetor as standard equipment but a three two-throat setup is available, primarily for the 375. The four-throat used on the 375 has larger venturis than the one on the 300.

Ford engineers approached the problem of designing a high-output engine from the correct direction by starting with its cylinder block. They made the block stiffer than the one for the standard engine to enable it to stand the high pressures that would be carried by its crankshaft. This was done by making the block's main bearing webs thicker and casting extra reinforcing ribs on them. All the block's oil passages were made larger to increase their flow capacity. The galleries necessary in the 300 hp block to keep its hydraulic lifters supplied with oil were blocked off in the 375 block because it is fitted with solid lifters.

It is common practice in most modern engines to make the oil pressure relief valve that limits the pressure in the engine's oil galleries an integral part of the oil pump. This places the relief valve at the source of the oil pressure. This has been satisfactory in thousands of engines but the Ford engineers working on the 375 got the idea, and it's a good one, that it might be better in an engine designed to run at high crankshaft speeds to control the pressure in its lubrication system from a point as far distant from the source of oil pressure as possible rather than at the source. To accomplish this a boss was cast on the rear end of the main oil gallery that runs the length of the cylinder block above the camshaft. This boss is drilled and threaded and fitted with a spring-loaded valve that opens when oil in the gallery exerts a pressure of 55 psi on it. When the valve opens, oil flows past it and returns to the oil pan's sump.



A later factory improvement for the 390-incher was this three-carburetor setup. It boosted the engine's power rating to 401.



Externally, the cylinder block for the 375 hp version of Ford's 390 is the same as the 300 block but internally it is stronger.

Oil pumps in 375 hp engines are still fitted with a pressure relief valve but this valve's purpose is to prevent the pressure in the full-flow oil filter from becoming so high when the oil is extremely cold that it will blow the filter's gasket. The valve is regulated to open when oil pressure reaches 105 pounds. Under normal conditions it will remain closed and all pressure regulation will be accomplished by the 55-pound valve.

A change made in the cylinder head capscrew holes in all 1961 large-displacement Ford engines is said to reduce cylinder block distortion when the capscrews are tightened. It consists of drilling the holes approximately ½-inch deeper and counterboring the upper ½-inch of their depth to a diameter larger than the threaded portion on the capscrews. The counterbore serves the same purpose as the chamfer around the upper ends of the capscrew holes in earlier blocks by preventing the metal around the holes from lifting and keeping the heads and gaskets from seating as they should. Placing the threaded portions of the holes deeper in the block makes the block stronger by putting more metal between the portion of the block that carries the load exerted by the capscrews and the block's top surface.

All 375 hp blocks are given a thorough inspection for cracks, weak spots, or other flaws that might cause trouble under the stresses of competition. Blocks that have any flaws in this category are rejected.

The same cast-iron crankshafts used in 300 hp engines are also used in 375 hp engines. Oil flow to their crankpins is increased more than one hundred percent by machining grooves around the circumference of their main bearing journals in line with the oil feed passages to the crankpins and by making the grooves in the bearing inserts larger. The main bearing inserts are conventional steel-backed type with copper-lead alloy bearing material. They are selected to provide a clearance of .0010 to .0031-inch.

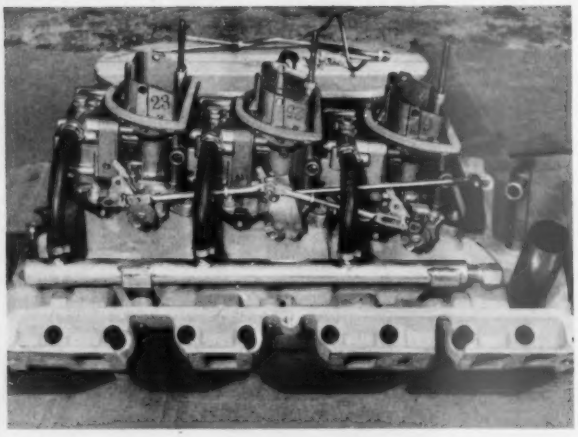
Connecting rods come off the same production line as those for 300 hp engines but they are subjected to a much more critical inspection. Those that are accepted are modified by narrowing their crankshaft bearing ends to provide a total side clearance of .014 to .024-inch when the rods are mounted on their crankpins. Bearing inserts for the rods are the same heavy-duty copper-lead type used for the main bearings. They are selected to provide a clearance of .0009 to .0029-inch.

Pistons are the same as those for 300 hp engines but after being selected for size to provide .0043-inch to .0049-inch clearance between the top of their skirt and the cylinder walls they are X-rayed for flaws. No flaws are allowed; pistons that don't pass the X-ray examination are discarded. Heads of the pistons have a concave shape to provide the required combustion chamber volume for the specified compression ratio. Tru-arc lock rings rather than conventional wire rings retain the full-floating piston pins in the rod and piston assemblies.

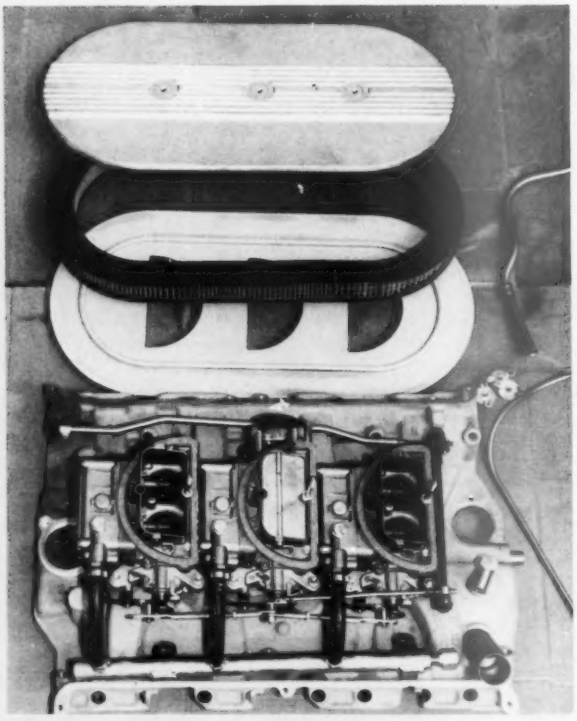
Cylinder heads for the 375 are the same as those on 1960 high-performance engines. They have 2.030-inch diameter intake valves that have 30 degree faces and 1.560-inch exhaust valves that have 45 degree faces. These valves are also used in 300 hp and 352 cubic inch engines. Combustion chambers in the heads are smaller than those in 300 hp heads to provide the higher compression ratio. It wasn't considered necessary to machine the chambers for volume control.

The camshaft in the 375 has the same grind as the shaft in 1960 high-performance engines. It actuates the valves through rocker arms that have a ratio of 1.76 to 1. Springs that close the valves are the same as the second production springs for 1960 high-performance engines. They exert pressures of 80 to 90 pounds when the valves are on their seats and 255 to 280 pounds when the valves are open. This pressure is said to be adequate for crankshaft speeds of 6400 to 6500 rpm. One change in this part of the engine is a switch from valve spring retainer washers machined from solid stock to stamped steel washers. The reason given for the switch is that failures experienced with the machined washers can be avoided through better control of the metallurgical content of the material possible with the steel used for the stamped washers.

An aluminum intake manifold with passages ten percent larger than those in the 300 hp manifold supports a four-throat carburetor that has a high air flow capacity. Because economy is of minor consequence in this engine, the intake



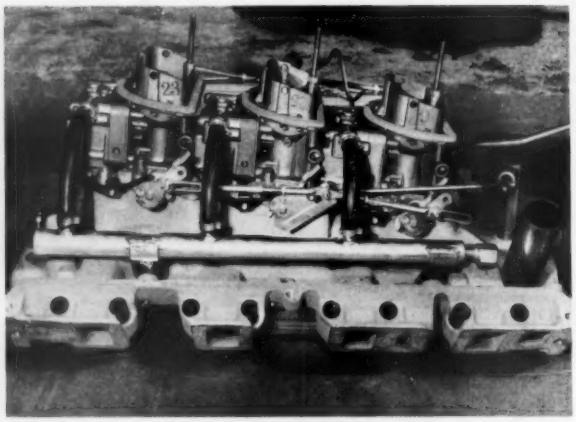
Here, the throttle valves in all three carburetors are fully opened. Progressive throttle linkage provides good engine performance at both slow and high car and engine speeds.



The three-carburetor setup includes a special aluminum intake manifold, special two-throat carburetors, progressive throttle linkage, a tubular fuel log and hoses, and a paper air cleaner.

Completely assembled, Ford's three-carburetor setup makes a neat, efficient looking package. When introduced, it was available only as an over-the-counter option.

The standard throttle linkage on Ford's three-carburetor setup is of the progressive type. For normal driving, only the middle carburetor is used. Here, the throttles are closed.

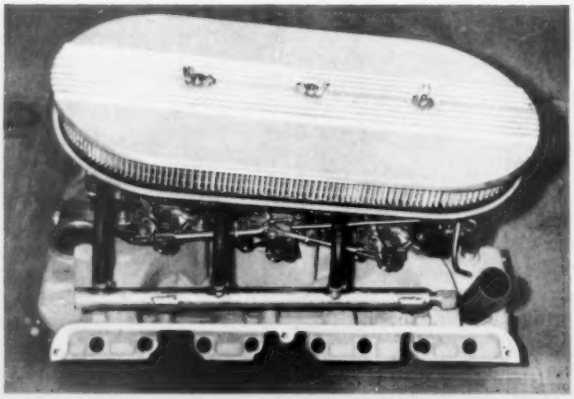


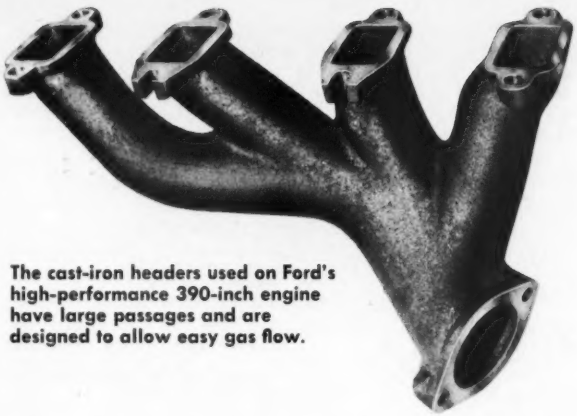
manifold does not have carburetor heat passages. Carburetor jets installed at the factory are for maximum performance rather than a compromise between maximum performance and maximum fuel mileage. The jets are said to be large enough for any type of competition.

The pump that delivers fuel to the carburetor has a higher pressure output than standard pumps; it doesn't have a vacuum diaphragm for windshield wiper operation. Between the pump and the carburetor is a truck-type fuel filter that has a paper filtering element. All fuel lines are ⅜-inch metal tubing. Air that flows to the carburetor passes through a special cleaner assembly that also has a paper filtering element.

The ignition distributor for the 375 hp engine has dual points to provide maximum coil saturation at high crankshaft speeds. It has governor weights that advance the timing as engine speed increases but it does not have a vacuum advance diaphragm for additional advance under light-load conditions. Spark plug wires have a steel conductor rather than the non-metallic conductor in the TVRS wire used on all other Ford engines. The ignition coil and resistor are the same as those for 300 hp engines.

Standard on the 375 are the cast-iron exhaust headers that were used on 1960 high-performance engines. In standard chassis these headers empty into a dual exhaust





The cast-iron headers used on Ford's high-performance 390-inch engine have large passages and are designed to allow easy gas flow.

system that consists of large-diameter pipes and low-restriction mufflers.

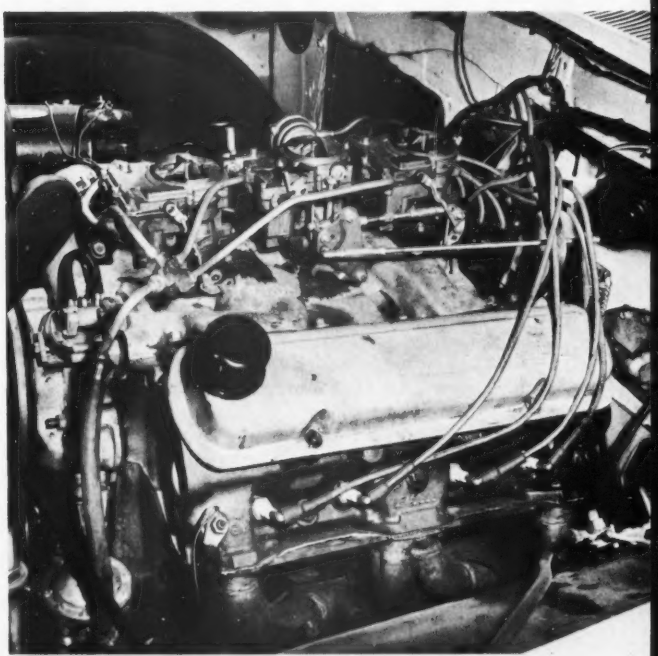
Inside the 375 water pump is a cast-iron impeller rather than one of the plastic impellers used in other engines. The cast-iron impeller has the same size, shape, and ability to move water as the plastic impellers but it was used because the plastic wasn't considered strong enough for the rough service the pump will get.

The generator on the 375 is similar to the 300 generator but its pulley is larger. The larger pulley reduces the generator armature's rotation speed in relation to crankshaft speed. This provides a safety factor against armature damage when the engine is running at high crankshaft speeds.

In February of '61 Ford announced a three-carburetor setup for the high-performance 390 cubic inch engine but these didn't get into the hands of hot rodders until several weeks later. The setup consists of three Holley two-throat carburetors on a special aluminum manifold. These carburetors provide a 40 percent greater air flow capacity than the original single four-throat. The setup was available only from parts departments of Ford dealers—it wasn't supplied as a factory option on cars delivered by the factory.

According to Ford, the new carburetion setup boosts the horsepower of the 390 high-performance engine to 401 at 6000 rpm and raises its maximum torque output to 430 pounds-feet at 3500 rpm. It is complete with a fuel line assembly, an air cleaner that consists of cast-aluminum housings and a paper element, and progressive throttle linkage. The throttle linkage is designed so that the engine runs on only the middle carburetor up to approximately one-third throttle and then it starts to open the end carburetors. Throttle valves in the rear carburetor start to open a few degrees of throttle movement before the valves in the front carburetor. The reason given for this is to add the extra venturi area of the end carburetors to that of the middle carburetor gradually rather than abruptly. It is said that opening the throttle valves in the end carburetors simultaneously could cause the engine to backfire. All the throttle valves reach their full-open position at the same time.

The carburetors have the same venturi diameters and are complete with the exception that those for the ends of the manifold don't have choke valves. They all have accelerating pumps and complete metering systems. Exhaust heat passages to the middle carburetor improve engine throttle response in low and medium engine speed ranges. This setup will fit all 332, 352, and 390 cubic inch Ford division engines.



This is one version of a Pontiac Super Stock engine. Pontiac has more high-performance equipment than any other automobile engine manufacturer. This engine has "Tri-power" carb.

Ray Nichels, the Pontiac specialist, built this Pontiac for a USAC stock car race. As engines for stock car racing cannot have multiple carburetors, this one has a single four-throat.

PONTIAC

Of all the automobile manufacturers, Pontiac is the most active in the optional high-performance field. They are exploiting to the fullest the advertising value of success in stock car racing events. They credit much of their ability to hold a position near the top of new car sales statistics to the number of wins their cars have recorded in the nation's biggest and fastest stock car races. Much of the credit for

these wins must go to the crews who prepare the cars and the men who drive them, but without Pontiac's cooperation in building and making available engine equipment that will develop the power necessary to push a modern passenger car at the speeds necessary for winning such races, and that will stay together, these men's efforts would be wasted.

Pontiac simplifies production problems by building their engine in only the one displacement size of 389 cubic inches but this simplification is nullified somewhat by the long list of special parts they make for the engine. Some of these parts increase power and torque outputs and others give the engine the ability to stay together during long periods of operation at high crankshaft speeds.

For 1961 Pontiac lists two basic engines, which are for their lowest priced cars, and at least eight additional engines of progressively higher horsepower ratings that are available as factory-installed equipment. Some of these additional engines are installed only in certain model cars but others are available in any model. Horsepower ratings of the engines in this group range from 215 to 348.

In addition to the standard production engines there are at least four more, as classified by valve timing and carburetion, that can be assembled from special high-performance parts obtainable only through the parts departments of Pontiac agencies. Factory horsepower ratings for these engines aren't available because they aren't listed as factory options but Ray Nichels, a Pontiac specialist in Highland, Indiana, who prepares Pontiacs for stock car racing, has pulled 363 horsepower from one of them set up with the optional four-throat carburetor and valve timing suitable for long, high-speed tracks.

Pontiac engineers started at the bottom and carried on from there when they decided to get into the high-performance field. From all appearances they wanted to make it possible for a fellow to build a maximum-performance Pontiac with Pontiac designed and manufactured parts he could buy over the counter at any Pontiac agency. If this was their aim, they succeeded.

The basic part of any engine is its cylinder block. Pontiac part catalogs list three of these: two for standard engines and one heavy-duty. Actually, there are only two blocks but because of an extra machining operation done on one of them the blocks must have three part numbers. Of the two standard blocks one is used for all but the two highest-rated engines and the other is for the Trophy 425-A high-performance options rated at 333 and 348 horsepower and available in any model car.

Both standard blocks have the same 4.06-inch cylinder diameters, the same oil galleries, and the same exterior dimensions and appearance. Where they differ is in their lower ends. The one for 425-A engines has special webs and caps for its number 2, 3, and 4 main bearing bores. Each of the caps is much heavier and stronger than the caps in the other block and is secured to the cylinder block with four capscrews instead of two. The four capscrews are so-positioned that their centers are in line.

The heavy-duty block is exactly the same as the four-bolt main bearing cap 425-A block except for a semi-circular chamfer machined in the upper end of each of its cylinders. The purpose of the chamfers is to match the block to optional heavy-duty cylinder heads.

All standard engines are equipped with a cast-iron crankshaft. The shaft has a stroke length of 3.75 inches, main bearing journals 3.00 inches in diameter, and crankpins 2.25 inches in diameter. A heavy-duty shaft, obtainable only from parts departments, is a steel forging, for greater strength. Its dimensions are exactly the same as those of the cast-iron shaft but extra attention was given to the fillets between its crank arms and its main bearing journals and crankpins to reduce the possibility of breakage at these points.

Two types of connecting rods are available. One of these is for all standard engines and the other is a heavy-duty over-the-counter option. Both types are forged steel and come off the same production line. Those that pass a very strict inspection are selected for heavy-duty use and given a special heat treatment that gives them the maximum strength possible with their material.

Caps on heavy-duty rods differ from the caps for standard rods by having 1/16-inch longer bolt bosses to make them stronger. Both rods have identical piston pin bore and crankpin bearing bore dimensions. They are designed for press-in pins.

Main and connecting rod bearing inserts for production engines are standard steel-backed babbitt except for the lower half of the number four main bearing. This is the bearing that controls crankshaft endplay. Its lower insert is a Moraine 400, which is a heavy-duty bearing.

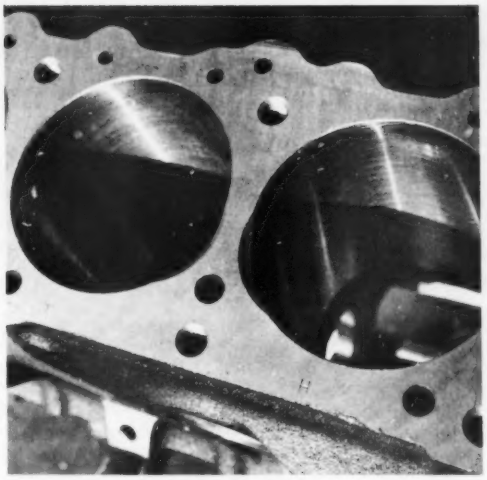
Two types of bearing inserts for both mains and rods are listed as heavy-duty options. These are Moraine 400, which consist of a steel backing that supports an aluminum matrix impregnated with lead alloy bearing material, and Clevite 77, which have copper-lead bearing surfaces. Both brands of inserts will give excellent service in any type of engine.

Pistons for all standard engines are cast aluminum alloy. Their slipper-type skirts are tin-plated to increase their resistance to scuffing. They have steel struts and cam ground skirts for expansion control. Their pin bores are offset .063-inch to reduce any tendency to be noisy under certain conditions. They are made with two different types of heads to provide the two compression ratios used in standard engines. Changing the shape of the pistons' heads made it possible to obtain different compression ratios with the same cylinder heads. Pistons for the lower ratio of 8.60 to 1 have concave heads. Those for the 10.25 to 1 ratio have flat heads that reduce the combustion chamber volume.

Reliefs are machined in the flat piston heads for valve head clearance. Reliefs of this type are usually to prevent the pistons' hitting the valves at the ends of exhaust strokes when the valves are floating but the reason given for the reliefs in Pontiac pistons is to provide clearance in the event that a valve should stick in its wide-open position for any reason. This apparently has happened in engines that have been idle for long periods of time. Valve float in Pontiac engines shouldn't provide any piston interference problems because the valve seats in the heads are so far from the piston heads. Four reliefs in each piston rather than the two required make it possible to use the same pistons in either cylinder bank. Because of their offset pin bores the pistons cannot be rotated 180 degrees, as would be necessary if they had only two reliefs, to change them from one cylinder bank to another.

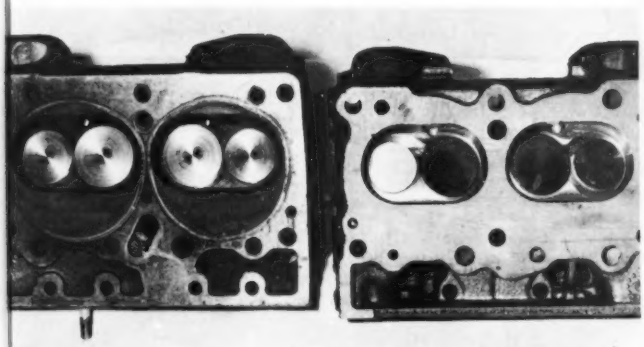
Heavy-duty pistons are available in only one type but they carry two part numbers because they are obtained from two manufacturers, which are ForgedTrue and Thompson Products. Both are forged aluminum alloy and have solid skirts and flat heads. As their heads don't have reliefs for valve clearance, these pistons effect a compression ratio increase of approximately .25 over that provided by standard flat head pistons. In other words, the compression ratio with heavy-duty pistons and standard cylinder heads would be 10.50 to 1 rather than the 10.25 to 1 with standard flat head pistons. All pistons use the same .980-inch diameter piston pins.

Heavy-duty lubrication system parts include an oil pump and oil pan. The pump is similar to standard pumps but it has a 60-pound relief valve instead of the 35 to 40 pound valve in the standard pump. To prevent the possibility of

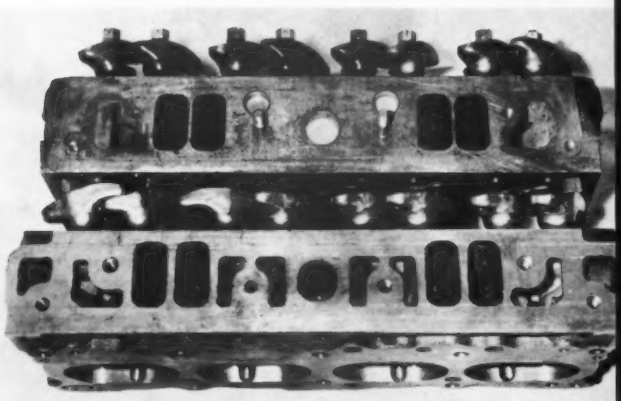


This is the top surface of a Pontiac heavy-duty cylinder block. This end of the block differs from standard blocks by having a small chamfer in the end of each of its cylinders.

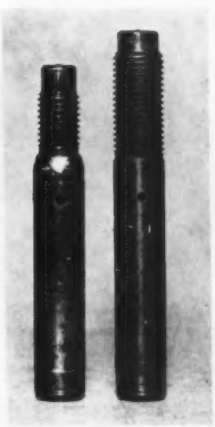
The lower end of Pontiac heavy-duty blocks differs from lower ends of standard blocks by having four-bolt caps on its number two, three, and four main bearing webs.



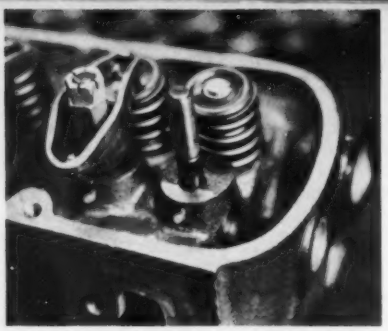
Combustion chambers in the heavy-duty head, at left, are longer on their intake valve side to increase the area around the valve heads when the valves are off their seats.

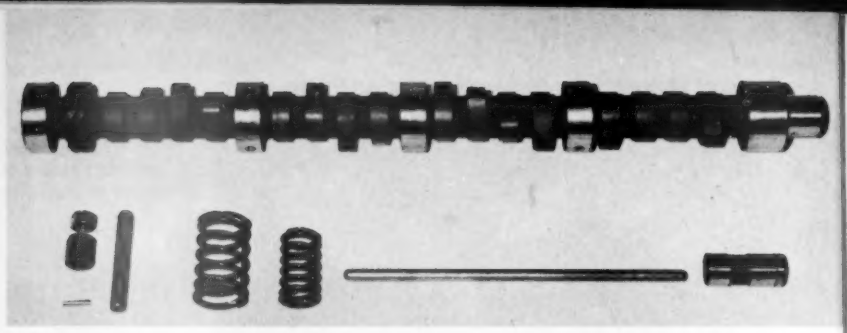


Camera angle makes these heads appear to be of different lengths but they are both Pontiac. The one at the top is the heavy-duty type for high-performance, other is standard.



At far left are one type of Pontiac heavy-duty forged aluminum pistons and a standard piston. The standard piston is the one with reliefs in its head. In the other photo are a standard rocker arm stud and a longer heavy-duty stud. Longer studs enable larger, double adjusting nuts to be used.





ABOVE—A Pontiac factory high-performance camshaft kit contains a special camshaft; heavy-duty rocker arm studs, pins, and nuts; dual valve springs; and special pushrods and lifters.

LEFT—It's common practice in Pontiac engines fitted with strong valve springs to pin the rocker arm studs in the cylinder heads to help prevent the studs' pulling out.

leakage between the lower end of the pump's gear housing and the housing's cover plate with the higher delivery pressure, the cover is made from thicker material than that used on standard pumps.

Instead of the floating-strainer oil pickup assembly on the standard pump, the heavy-duty pump has a stationary pickup tube. The end of the tube is located near the right side of the pan's sump. This is where the oil tries to collect when a car is in fast left turns.

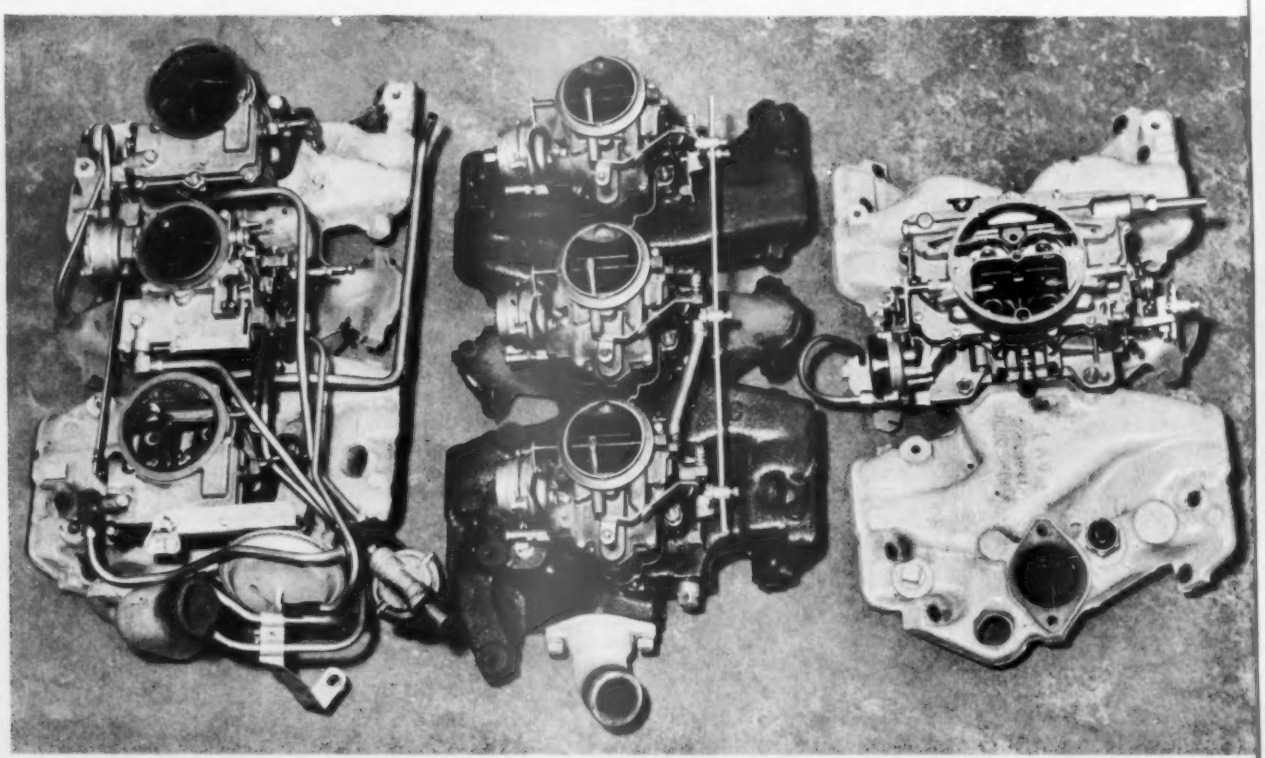
Standard oil pans have a four-quart capacity. In the heavy-duty pan this is doubled by a larger sump. The pan is well-baffled to control movement of the oil during rapid acceleration and in fast turns. Additional oil capacity not only reduces the possibility of the oil supply's being depleted during a long race but it also allows the oil to run a few degrees cooler. As oil carries away much of the heat created in an engine's crankshaft bearings, keeping it cooler can lengthen bearing life.

Cylinder heads with three part numbers are listed but two of these, which are for standard engines, are identical except for the depth of their combustion chambers. Chambers in one are approximately .040-inch shallower than those in the other to effect a compression ratio increase. With standard flat-head pistons, the compression ratio with cylinder heads that have the deeper chambers is 10.25 to 1; with the shallower chambers it is 10.75 to 1. Combustion chambers in all the heads are fully-machined. This insures maximum control over the chambers' volumes. When combustion chamber volumes vary, compression ratios also vary.

The design of standard cylinder heads for 1961 engines was changed to reduce the heads' weight. These changes, because of their effect on water passages and other design features, made it necessary to also redesign the intake manifold. Heavy-duty heads have the same basic design as 1960 and earlier heads.

In their features that affect breathing and compression ratio the heavy-duty heads are considerably different than standard heads. Their combustion chambers are shallower

These three different carburetor and manifold setups were used by Edelbrock Equipment Co. for a series of dynamometer tests they ran with a Pontiac engine. The one at the left is a Tri-power with three Rochester carburetors and standard vacuum-actuated secondary throttles, the middle one has three Stromberg's with synchronized throttles, and the other has a four-throat.



to provide a higher compression ratio. They have 1.920-inch diameter intake valves compared to the 1.880-inch standard valves. Their exhaust valves are 1.660-inch in diameter compared to the standard 1.600-inch valves. The difference in valve head diameters isn't great but by enlarging the valve seats to make them contact the upper edges of the valves' faces it was possible to enlarge the valve ports approximately .100-inch. This gives an appreciable increase in valve port area and in the engine's breathing ability. Seats for the intake valves have an angle of 30 degrees and the angle of exhaust seats is 45 degrees. To match the greater flow capacities of the larger valve ports, intake and exhaust ports and passages were made larger than those in standard heads.

Oversize valves aren't of much value unless there is adequate area in the combustion chambers to let fresh mixture and exhaust gases flow freely past their heads when the heads are off their seats. To provide this area, combustion chambers in the heavy-duty heads were enlarged around the intake valve seats by making them longer. This caused the chambers to extend past the cylinder openings in the block, which is the reason for the chamfer machined in the upper end of the cylinders in heavy-duty blocks. Without the chamfer, part of the fresh mixture that flows past the intake valves would strike the right-angle obstruction formed by the area of the cylinder block that extends into the combustion chamber. Chamfering the block in this area improves this condition somewhat by making the obstruction less abrupt. The obstruction still exists but at least it is angled in the direction of mixture flow.

Heavy-duty cylinder heads can be used on standard cylinder blocks after the upper ends of the cylinders have been chamfered. Directions for doing this are supplied with the heads. The job can be done quite easily with small grinding wheels.

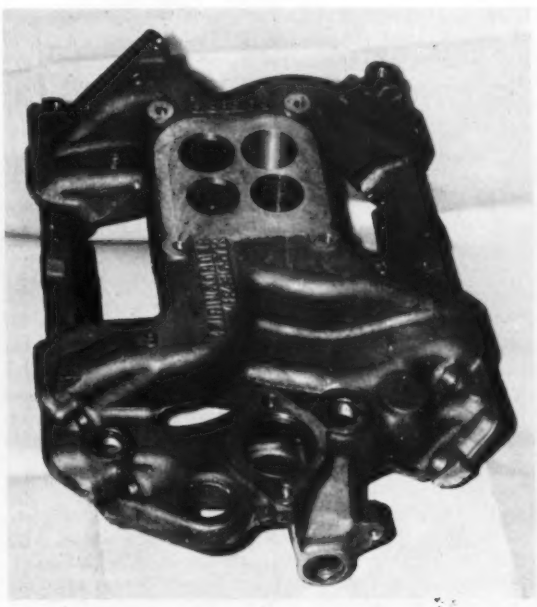
Included with heavy-duty heads are special rocker arms similar in design to standard arms but that are heat-treated to give them maximum strength and that have a ratio of 1.65 to 1 rather than the standard 1.50 to 1. Because of the arms' different ratio, the studs in the cylinder heads that support them are .050-inch farther from the valve guides than they are in standard heads. Also, the studs are longer than those in standard heads. Included with the rocker arms are special fulcrum balls. Because of the different stud location they require, it's obvious that the high-ratio rocker arms can't be used on standard cylinder heads.

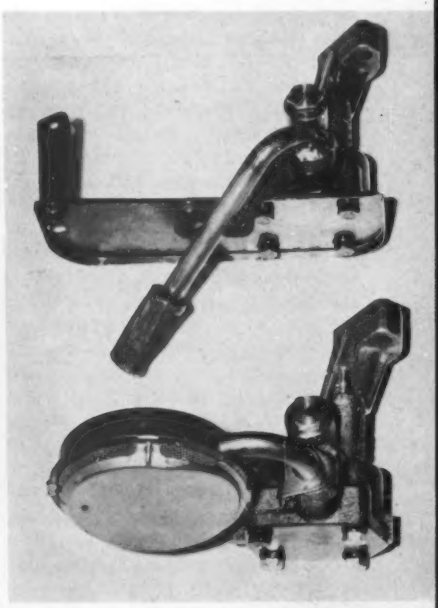
Also supplied with the heads are special restrictor-type plugs for their oil passages. These limit the quantity of oil that flows to the rocker arms to prevent the valve assemblies from becoming flooded when the engine is operated at high crankshaft speeds for long periods of time. The heavy-duty heads also require special rocker arm covers.

All standard engines have camshafts ground for hydraulic lifters. One of these is a semi high-performance shaft that is standard equipment in engines equipped with Tri-power carburetion setups. Special hydraulic lifters designed for higher crankshaft speeds than those possible with standard lifters are used with it. This camshaft and lifter combination is said to be good for engine speeds up to 5200 rpm, where the valves start to float.

High-performance camshafts with two different grinds are available from parts departments. They use solid-lifters and can be installed in any cylinder block; however, unless the engine has heavy-duty heads the rocker arm studs in the heads will have to be replaced with longer studs. The purpose of the longer studs is to permit the installation of special rocker arm fulcrum ball nuts and lock nuts.

The high-performance camshafts are identified by the name McKeller and the numbers 7 and 8. McKeller is the name of the Pontiac engineer who designed the cams. The number 7 grind is for acceleration and is good for street, drag racing, and other applications where good acceleration





The intake manifold at the right is the Pontiac heavy-duty four-throat model. It isn't exhaust heated, which makes it impractical for normal driving. At far right, above, is a heavy-duty oil pump. This pump does not have a floating pickup.

characteristics are desirable. The number 8 grind is for high-speed operation, such as at the lakes or Bonneville, or on the tracks at Daytona and Darlington.

With a McKeller grind, standard valve springs can't control the valves as they should. Special inner and outer springs Pontiac supplies with the shafts must be installed. Pressures exerted by the dual springs are much greater than those of the standard single springs. The valve stem oil shields on standard engines are not used with dual springs, and standard spring retainer washers must be replaced with special heavy-duty washers. Special pushrods supplied with McKeller shafts are slightly longer than standard rods.

When a McKeller shaft is installed in an engine that has standard cylinder heads it is a good idea to replace the engine's valves with special Pontiac valves that have the same head diameters but are made of better material, that have contours that improve fresh mixture flow into the cylinders and the flow of exhaust gases out of the cylinders, and are polished. It would be a good idea when installing these valves to enlarge the valve ports as much as possible.

Intake manifolds for standard engines are made in three types. These are all cast-iron. One is for a single two-throat carburetor, another is for a single four-throat, and the third is for three two-throats. The three two-throat manifold is for the standard Tri-power setup.

There are also three high-performance manifolds. One is for standard cylinder heads. It is made of cast iron and for a four-throat carburetor. It has larger passages than those in the standard four-throat manifold and it uses a carburetor that has larger venturis than those in the standard carburetor. It has exhaust heat passages around its risers.

The other two high-performance manifolds can be used only with high-performance heads. Both are aluminum and neither is exhaust heated. Their passages are large to reduce to the practical minimum the resistance they present to the fuel and air mixture that flows through them. One is for the same four-throat carburetor used on the cast iron four-throat manifold and the other uses three standard Tri-power carburetors.

A definite improvement for all Tri-power setups is a mechanical progressive throttle linkage Pontiac supplies to replace the standard vacuum mechanism that actuates the throttle valves in the end carburetors. The disadvantage of the vacuum mechanism is that the throttle valves it controls are either fully-closed or fully-opened—there isn't any such thing as partial opening. With the mechanical linkage the valves in the end carburetors start to open when the middle carburetor's throttle shaft reaches a predetermined position. Then, as movement of the throttle linkage is continued by moving the accelerator pedal, the end carburetors continue to open so that the throttle valves in all three carburetors reach their full-open position at the same time. Included in the mechanical linkage kit are special throttle shafts for the middle and rear carburetors and the linkage to connect the throttle shafts of all three carburetors.

All Tri-power carburetors are Rochester's and all four-throats are Carter's. The same end carburetors are used for all Tri-power setups but the middle carburetor, which has smaller venturis, differs according to the engine and transmission combination on which it will be used. The high-performance Carter four-throat has ⅛-inch larger primary bores than the standard four-throat. Three different four-throats are used on standard engines.

Other high-performance options for the carburetion system are low-restriction air cleaners for either four-throat or Tri-power carburetors. These have paper filtering elements. Another item is a special fuel pump that delivers fuel at a greater pressure than standard pumps.

Standard ignition systems have a single-point distributor that has both centrifugal and vacuum advance mechanisms. A high-performance option is a dual-point distributor that has a centrifugal advance mechanism only. The advance mechanism in this distributor has a greater range than the centrifugal mechanism in the standard distributor to give the ignition timing the necessary lead the engine requires at high crankshaft speeds. Included with the dual-point distributor are a special coil, coil resistor, and lead wire.

Spark plugs that are standard in production engines are AC 45 S. For high-performance street use Pontiac recommends AC 43 plugs, or the equivalent in another make, and for drag racing, AC 42 or equivalent.

Two types of exhaust manifolds are supplied on standard engines. One of these is of the conventional collector type that has a branch for each of the cylinders in the cylinder bank. The other, which is used on 425-A engines, is best described as a "W" type. It has larger passages, longer branches, and bends of longer radius.

Available over the counter are manifolds that are actually cast-iron headers. Each of these consists of two Y-shaped branches that are not connected internally in any way. Each of the Y's accommodates two cylinders whose firing impulses are so-spaced that the pressure of exhaust gases from one will not affect the flow of exhaust from the other. Because of their design each manifold has two outlets but the outlets are in a common flange. These manifolds weigh approximately twice as much as the W-type used on 425-A engines.

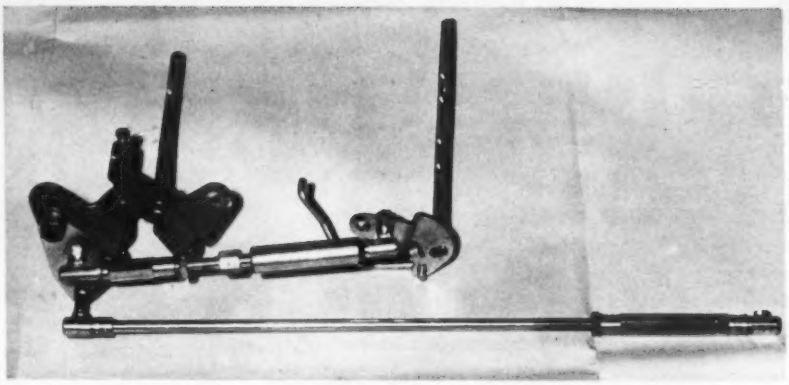
Because of the restrictions in standard Pontiac chassis, engines that have the high-performance cast-iron headers must also be fitted with a special starting motor. This motor differs from standard motors by having its solenoid on the bottom rather than the top and a different pinion gear housing. Also, different clutch linkage must be installed to clear the headers.

For engines to be used for racing, Pontiac has a lightweight flywheel and heavy-duty clutch assembly. A standard clutch cannot be mounted on this flywheel, and six special bolts are required to secure it to the engine's crankshaft. Included in the clutch assembly are a pressure plate assembly, a driven disc, and a special fulcrum ball for the release bearing arm.

A special heavy-duty harmonic balancer assembly is available in two types for standard engines that are to be driven hard. One of these has a single-groove pulley for cars that have standard steering gears and the other has a double-groove pulley for cars that have power steering.

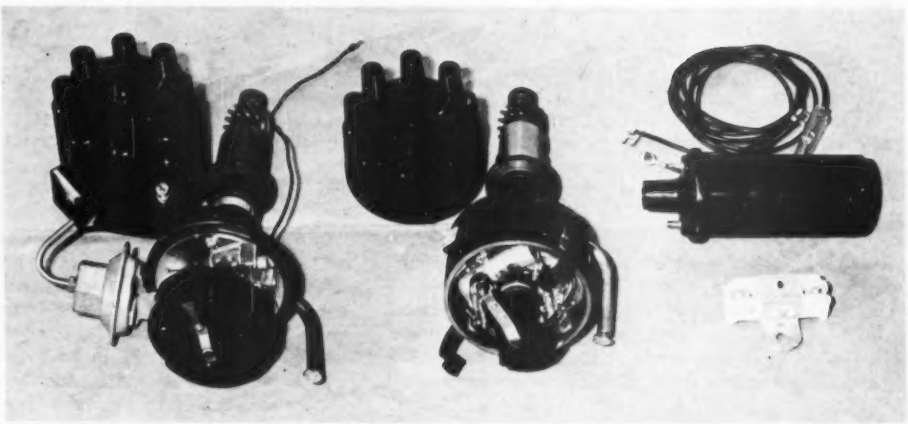
For engines to be used for racing there is another heavy-duty harmonic balancer-pulley assembly that has a pulley with extra-deep grooves. The purpose of the deeper grooves is to help prevent belt throwing and turning over at high crankshaft speeds. Included with this assembly are water pump and generator pulleys that have similar grooves. The generator pulley is larger than the standard pulley to reduce the speed at which the generator's armature rotates in relation to crankshaft speed. Slower armature speed lowers the centrifugal force that acts on the armature's parts. It's this force that damages armatures. Also included is a belt of the correct length for the new pulley setup.

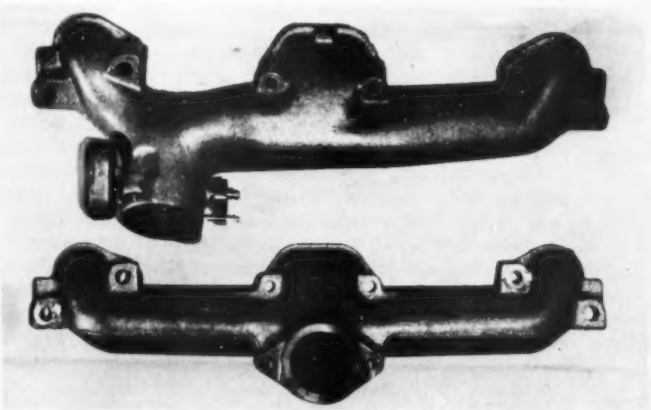
There are two optional fans. One has standard blades but a clutch-type drive that slips at high rotational speeds to limit the blades' speed. This reduces the power required to rotate the blades. The other is driven in the conventional manner but its blades are not twisted as much as those on standard fans. The blades don't move as much air per revolution as those on a standard fan but less power is required to drive them at any given engine speed. **DON FRANCISCO**

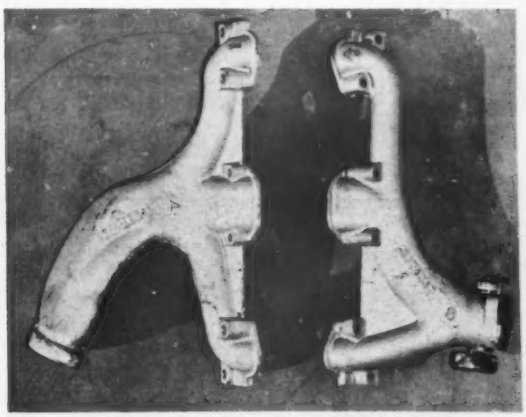


Pontiac supplies this kit to replace the vacuum actuating mechanism for the end carburetors on Tri-power setups. Parts are for front and middle carbs.

At the left is the standard Pontiac ignition distributor for street engines, and beside it is the special high-performance distributor that has dual breaker points but does not have a vacuum advance unit. The coil and resistor are used with the special distributor.







Above and at the right are the three kinds of exhaust manifolds available for Pontiac engines. Above is the standard type that is used for normal production. Above, right, are the manifolds that are standard on production "A" series engines. These are considerably better than the normal production type. At right are the over-the-counter heavy-duty manifolds that are really cast-iron headers. Each of these has two individual sections, each of which handles two cylinders. These are quite heavy.

GETTING THE MOST OUT OF A SUPER STOCK

THE NECESSARY parts required for a high-performance engine form only one of the ingredients for a powerplant that will deliver maximum power from each cubic inch of its displacement. Another ingredient, equal in importance to the necessary parts, is correct assembling of the parts. Correct assembling includes bringing certain of the engine's specifications up to factory engineering standards, aligning the engine's moving parts correctly, and establishing adequate but not excessive clearances between the parts.

Some of the steps required when assembling a high-performance engine would not be necessary if it weren't for the "tolerances" required by mass production techniques. Because it is practically impossible for the dimensions of engine parts manufactured as rapidly as they are in modern automobile plants to be exact, the dimensions are allowed to be a little oversize or a little undersize. The allowed variation in a dimension is the dimension's tolerance. A part is usable if its various dimensions fall within their tolerances.

In an engine being built to deliver maximum performance but still be considered stock close attention must be given to many of its dimensions and specifications. The most important of these are those that determine the engine's compression ratio. In most instances dimensions and specifications that have an influence on compression ratio must be the minimum allowed by their tolerances so the ratio will be as high as possible.

Checking each of the parts that must be checked and making the necessary corrections to change conditions in an engine to what they should be lengthens the time required to assemble the engine. Some of the parts must be assembled temporarily for the check and then disassembled so that the corrections can be made. This additional time and work are things that can't be avoided if the results you want are to be enjoyed. The job can be simplified as much as possible by taking the necessary steps in their logical order. The crankshaft makes a good starting point.

Things to check on a crankshaft are the length of each of its throws, the placement of its crankpins in relation to each other, and the clearances between its bearing surfaces and the bearings that will be used on them. Accurate determinations of throw length and crankpin placement require special equipment usually found only in the better crankshaft regrinding shops.

Crank throw length determines the length of the stroke through which the pistons connected to each of the shaft's crankpins will move. Any stroke variation between cylinders will cause each cylinder to have a different compression ratio. A longer stroke will raise the ratio and a shorter one will lower it. The stroke shouldn't be shorter than specified and a good cheater might get by with one a few thousandths of an inch longer, but whether just right or a hair long, the stroke length for all cylinders should be the same.

The crankpins should be spaced exactly 90 degrees apart around the circumference described by their centers when the shaft rotates on its main bearings. Any variations in this placement will cause the pistons connected to crankpins that aren't correctly positioned to reach top center in their cylinders before or after they should in relation to valve timing. If the error were great enough the power output of the cylinders involved could be affected.

Any errors found in the crank throw length or crankpin placement on a modern crankshaft should be small ones. These can be corrected by grinding the crankpins slightly undersize to reposition their centers. Pin diameter can be reduced as much as .030-inch for this purpose without affecting a crankshaft's durability in any way. The rare crankshaft that can't be corrected in this manner should be replaced.

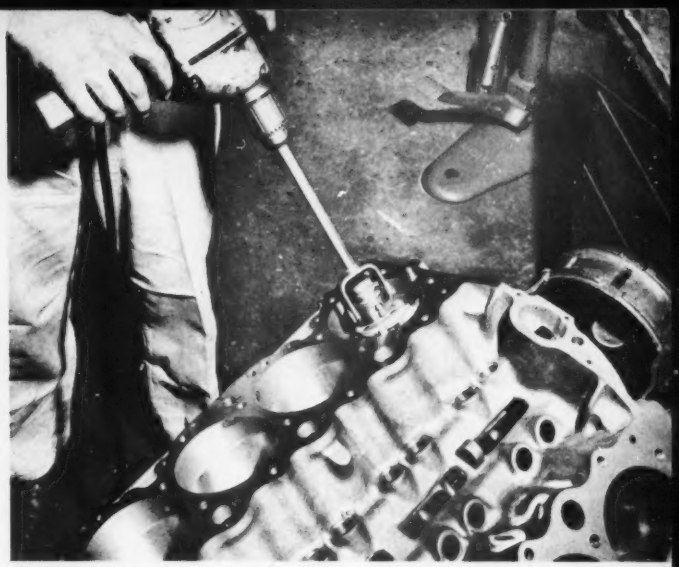
A shaft's main bearing journals and crankpins should be Micro-finished to make them as smooth as possible. This also is a job for a crankshaft grinding shop. It requires the shaft to be rotated in a special machine while an abrasive belt, which is also rotating, is held against each of its bearing surfaces in turn. Only a fraction of a thousandth of an inch of material is removed by Micro-finishing to polish bearing surfaces but additional material can be removed in this manner to provide the desired clearances between the surfaces and the bearings that will be used on them; however, material cannot be removed to increase bearing clearances until the bearings have been installed on their respective journals and crankpins and their clearance determined.

Each of the connecting rods should have its correct center to center length, which is the distance from the center of its piston pin bore to the center of its crankshaft bearing bore, have the correct crankshaft bearing bore diameter, provide the correct fit for the piston pin that will be used in it, and be correctly aligned.

As far as their center to center length and connecting rod bearing bore diameter are concerned probably the best thing to do with the rods is to take them to a shop that rebuilds rods. There's usually one or two of these in every city of any size. They have the equipment necessary to make the measurements and corrections.

A rod's center to center length is changed by moving the center of its crankshaft bearing bore in the necessary direction. This is done by grinding layers of material from the mating surfaces between the rod and its cap to make the bore oblong rather than round and then remachining the bore to make it round again and so its center is the correct distance from the center of the piston pin bore. A rod longer than it should be will raise the compression ratio of its cylinder and one shorter will lower the ratio. The diameter of the crankpin bearing bore is important because it determines the diameter of the bearing inserts installed in it. This, in turn, determines the clearances between the

The correct deck clearance, necessary for a high compression ratio, is established by grinding the top of the cylinder block.



Piston to cylinder wall clearance is increased to approximately .005-inch with a cylinder hone to reduced piston to wall drag.

inserts and the crankshaft's crankpins.

Piston pin bores in the rods for many modern engines are slightly smaller in diameter than the pins so that the pins have to be pressed into them. Pin bores of this type don't require any special attention unless they are to be reworked for floating pins. This is usually done by honing them to a diameter approximately .0009-inch larger than the pins. The material of the rod is used as the bearing surface. Some fellows go to the trouble of installing bronze bushings in the bores but this requires the removal of quite a bit of material unless special undersize pins are used. Undersize pins involve special pistons which might not be considered as stock equipment by drag racing associations. When floating pins are used in stock pistons used originally with pressed-in pins the pistons must be reworked by machining grooves in the outer ends of their pin bores for lock rings.

Pin bores in rods designed for floating pins should be honed to a diameter slightly larger than that of the pins. The specified clearance for such installations is usually .0003 to .0005-inch.

Alignment of the rods should be checked after all machine work has been done on their bores. The theory of connecting rod alignment is to make center lines through the bores parallel when viewed from any direction. Any misalignment in its center-lines when a rod is viewed from the side means that the rod is bent. Misalignment in the center lines when the rod is viewed from the top means that the rod is twisted. Any deviation in alignment should be slight and easily corrected by bending or twisting the rod in the correct direction. A rod drastically out of alignment for any reason should be replaced rather than corrected and used.

After the crankshaft and connecting rods have been brought up to specifications their bearing clearances can be determined and the necessary corrections made. For the average hot rodder the easiest and most practical way of

One of the most important parts of any engine assembly job is to clean the cylinder block and the other parts, inside and out.

Bearing clearances of any modern engine depend on careful tightening of all main bearing capscrews and connecting rod cap nuts with a good torque wrench.

checking the clearances is with Plastigage, a product of Perfect Circle Corporation. To do this, install the main bearing inserts and the crankshaft in the cylinder block and the connecting rods on their crankpins. It is important that the crankshaft be installed in the engine exactly as it will be for the final assembly, with the exception that main bearing seals aren't required. Bearing bores in the block and caps must be perfectly clean, both sides of the inserts must be clean, and the capscrews that secure the caps to the block must be tightened to their specified torque. The block must be supported so that its crankshaft side is up.

Starting with the number one main bearing, remove its cap and place a piece of Plastigage equal in length to the width of the bearing insert in the cap on the crankshaft's journal. Reinstall the cap and torque its capscrews to the specified tension. Be careful not to rotate the crankshaft while the Plastigage is in place. Remove the cap and compare the width of the flattened Plastigage with the graduations on the side of the Plastigage envelope. The graduation that corresponds with the Plastigage's width is the clearance between the bearing and its journal. Repeat this procedure for the rest of the main bearings.

Plastigage is used for connecting rod bearings in the same manner described for main bearings. Place a length of it on the crankpin, install the rod's cap and bearing assembly, torque the cap nuts to the specified tension, and then remove the cap. Determine the bearing's clearance with the graduations on the envelope.

Suitable clearances for both main and connecting rod bearings for late-model high-performance engines seem to be fairly well standardized at .0025 to .0030-inch. It isn't likely that clearances greater than .0030-inch will be found but the easiest way to reduce those that are is by installing inserts that are slightly thicker than those that were checked. These may or may not be obtainable from the engine's manufacturer.

In the interest of satisfactory bearing life don't accept any bearings that aren't the heavy-duty types recommended by the engine's manufacturer. If it isn't possible to reduce the clearance to .0030-inch, or possibly slightly more, by installing thicker inserts the only alternatives are to have the crankshaft's bearing surfaces ground to the next standard undersize or replace the shaft. A crankshaft used for a replacement should be given the same inspection and correction treatment specified for the original shaft.

Bearing clearances are nearly always less than the .0025-inch minimum specified. This gives a fellow two alternatives. One is to install thinner bearing inserts, if they are available. The other is to use the inserts that were checked and have the crankshaft's bearing journals Micro-finished to diameters that will provide the desired clearance. One method is as suitable as the other but using different inserts is usually the easiest.

After establishing the correct main bearing clearances check the crankshaft's endplay in the cylinder block. Before doing this be sure the thrust bearing cap was installed according to factory recommendations. Incorrect cap installation can cause the endplay to be less than it would be if the cap had been installed correctly. The only way to determine how the cap should be installed is by consulting a shop manual for the engine.

The easiest way of measuring endplay is by forcing the crankshaft forward so that one of its surfaces that controls the endplay is seated firmly against the flange on the engine's thrust bearing and then measuring the distance between the shaft's other thrust surface and the bearing's other flange with the blade of a thickness gauge. The endplay should be within the tolerance specified by the engine's manufacturer. Endplay less than specified can be increased by having one or both of the bearing's thrust flanges machined in a bearing sizing machine. Most auto parts stores that have machine shops are equipped to do this sort of work. Clearance greater than that specified can be reduced by installing inserts that have thicker thrust flanges.

While checking connecting rod bearing clearances also measure the total endplay of each set of connecting rods on their crankpin. This can be measured with a thickness gauge blade. Push both rods tightly together so that one is forced against the flange at the end of the crankpin and then insert the thickness gauge between the flange at the other end of the pin and the connecting rod adjacent to it. Rod endplay specifications vary between engines but a good tolerance for a high-performance engine is .015 to .020-inch.

Adequate rod endplay is necessary primarily to allow sufficient quantities of oil to flow through the rod bearings. Oil cools as well as lubricates the surfaces it contacts but it cannot fulfill its cooling duties unless enough of it is allowed to flow through a bearing. In a high-performance engine this cooling is more important than it would be in an engine used only for normal driving. The endplay can be increased by machining the sides of the connecting rod big ends. As the amount of material that would have to be removed for this purpose would be only a few thousandths of an inch, the machining must be done carefully to prevent the endplay's exceeding its tolerance.

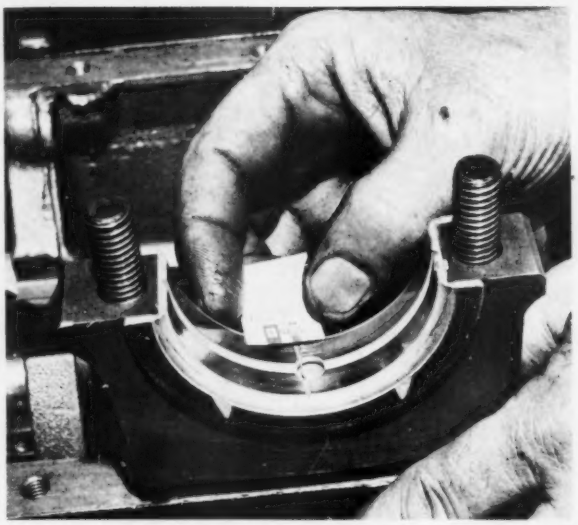
While the crankshaft and rod and piston assemblies are in the block, measure the deck clearance for each of the cylinders. Deck clearance is the distance from a piston's head to the top surface of the cylinder block when the piston is at top center in its cylinder. It is an important measurement because it has an influence on combustion chamber volume, which is a factor that helps determine compression ratio. As deck clearance becomes greater, combustion chamber volume becomes greater and the compression ratio becomes lower.

All engines have a factory tolerance for deck clearance. For the engine to have its maximum possible compression ratio the clearance must be the minimum allowed by the tolerance, and for the engine to retain its stock status the clearance must not be less than the minimum.

Deck clearance is determined by rotating the crankshaft to place the piston to be checked at top center in its cylinder and then measuring from the top of the piston to the top surface of the cylinder block with a depth micrometer or some other precision measuring device. Unless a crankshaft's stroke is longer than it should be, or connecting rods have a longer center to center length than they should have, or the compression height of the pistons is greater than it should be, deck clearance will nearly always be greater than the factory tolerance minimum. To reduce it to the minimum, have the top surface of the cylinder banks ground or milled to remove the required amount of material. This is another job many auto parts machine shops can do, and regular machine shops usually have milling machines big enough to accept a cylinder block. The new surface should form the same angle to the cylinders as the original surface.

An engine's crankshaft, connecting rods, pistons, and piston pins should be balanced as an assembly after all machine work has been done on them. If all these parts are strictly stock, meaning that they weren't remachined in any way that would change their weight, and from the same engine, they should be usable without rebalancing. But if anything was done to them to change their weight, it is absolutely necessary that they be balanced if the engine is to run satisfactorily.

Balancing is another of those specialized jobs that a rodder must have done. Fortunately, there are many com-

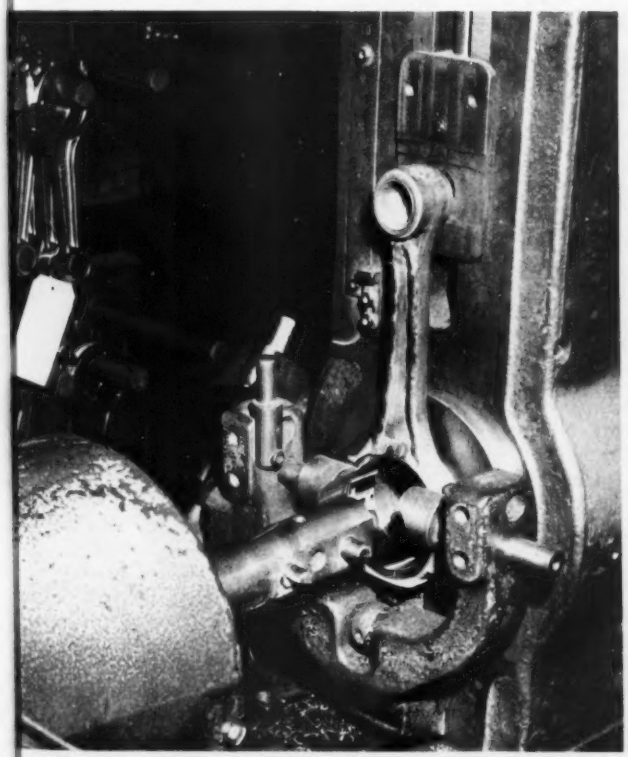


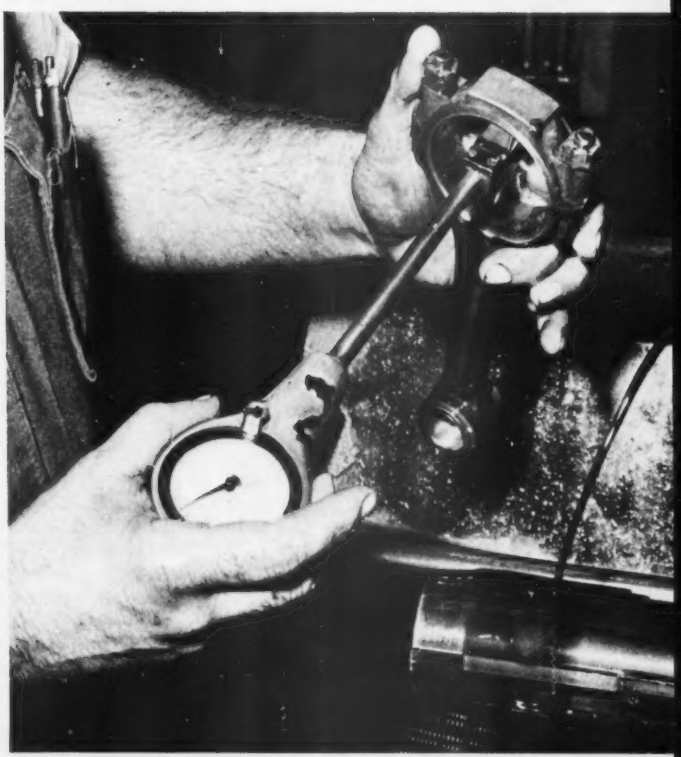
ABOVE—One easy way to check the clearance of main and connecting rod bearings is with Plastigage, as shown here.

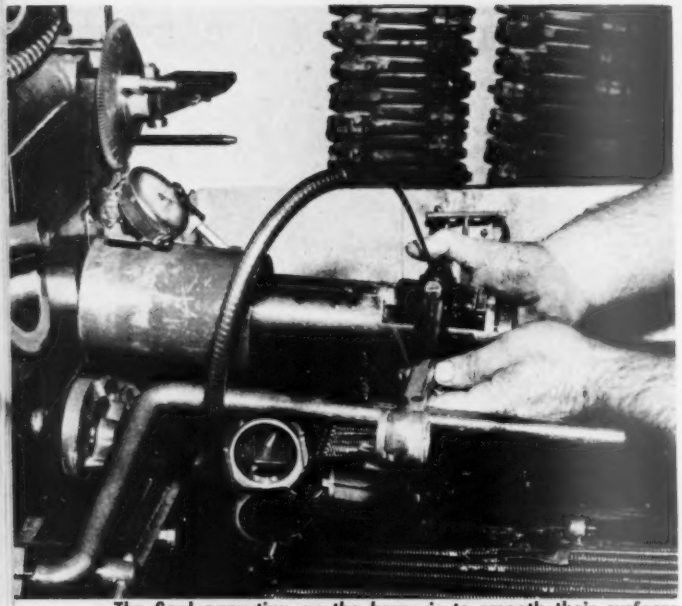
RIGHT—Correct crankshaft endplay is important to free rotation of crankshaft. It can be checked with a dial indicator.

LOWER RIGHT—To check connecting rod big-end bores for diameter and shape use a special dial indicator of this type.

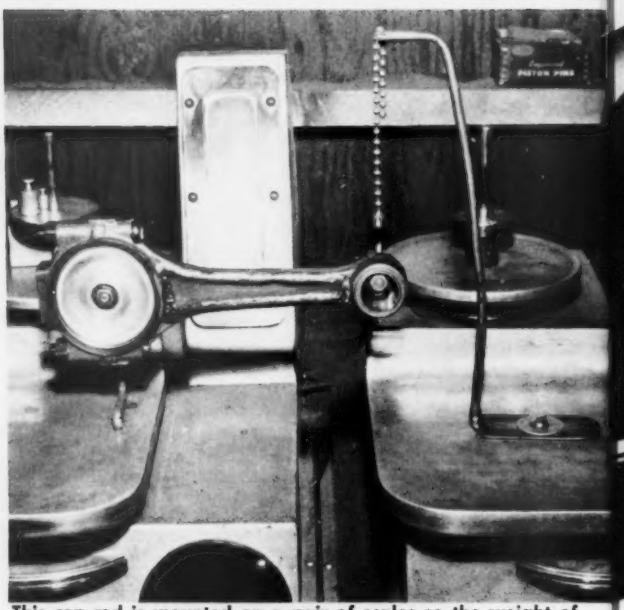
BELOW—Rod big end bores are made round and correct diameter by machining their surfaces in a special boring machine.







The final operation on the bores is to smooth their surfaces and finish them to their specified diameter with a wet hone.



This con rod is mounted on a pair of scales so the weight of its ends and its total weight can be checked at the same time.

panies throughout the country that have the equipment and the ability to turn out satisfactory balancing jobs. You may have to ship your parts to one of these if you don't live within easy driving distance of one of them but this is one of those things an engine builder learns to expect.

During the balancing process material is removed from the pistons to bring the weight of all of them down to that of the lightest and the weight of the rods is reduced to make all their big ends the same, all their small ends the same, and their total weight the same. Then weight is either added to or removed from the crankshaft to balance it in relation to piston and rod weights. The only part of these operations that can create any problem is removing weight from the connecting rods. Rods have balancing bosses at each end for this purpose. The difficulty is that sometimes grinding material from the boss on the pin bore end causes the bore to distort. If this is done after the pins have been fitted, the bores must be honed again to make them round and of the correct diameter. For this reason it is usually better to have the parts balanced before fitting the pins. This adds complications to the engine reworking job but it cannot be helped.

To guarantee minimum friction between the pistons and their cylinder walls, have the cylinders honed oversize the amount necessary to give each piston a skirt clearance of .005-inch. This clearance seems to be adequate in all late model engines. Each piston should be measured individually and the cylinder in which it will be used enlarged accordingly.

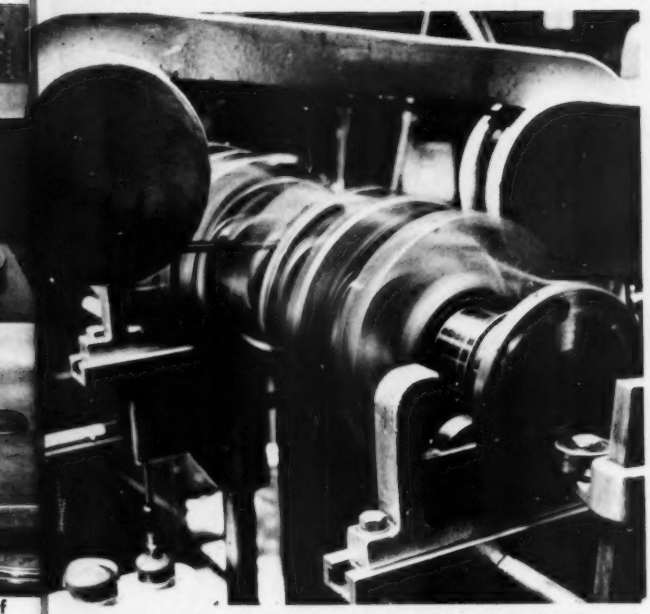
A very important part of honing any cylinder block is to thoroughly clean the block after the honing has been finished. One of the most practical methods of cleaning the cylinder walls is with soapsuds and lots of water. Wash each of the walls thoroughly with a clean cloth saturated with the soapsuds and then flush the cylinders with plenty of clean water, preferably from a garden hose. Residue from the honing that worked its way into the lower end of the crankcase can also be washed out with soapsuds, possibly followed by a scrubbing with solvent or kerosene to remove residue caught in oil or other matter that wasn't dissolved by the soapy water.

Proper cleaning of the cylinder block is one of the most important steps in any engine assembly job. Grit deposited in cylinder walls by a hone will ruin a set of piston rings in only a few minutes running time. Grit in the crankcase can be picked up by the lubricating oil and circulated through the engine's bearings. A full-flow oil filter will capture most of the grit but a certain percentage of it could get through to the bearings. The result would be very short bearing and crankshaft life. Don't take a chance on this happening in your engine. Spend another hour or so and be sure that all the block's surfaces, both inside and outside, are spotless before starting to assemble the engine.

When intalling the pistons on their respective connecting rods be sure their numbers are the same as those on the rods and that each piston is positioned correctly on its rod. Connecting rods must be installed on their respective crankshaft crankpins so their numbers are on a specified side of the shaft and pistons must be installed in their cylinders so their various design features are in the correct relationship to the cylinder heads and to the direction of crankshaft rotation. Be sure that lock rings for floating piston pins are correctly installed and that none of them are left out, and that pressed-in pins are centered in their connecting rods.

Before installing the piston rings on the pistons check each ring in the cylinder in which it will be used to determine whether it has adequate end gap. This is done by inserting a ring in the cylinder so it is square with the cylinder's wall and then inserting the blade of a thickness gauge between its ends. The end gap should not, under any circumstance, be less than that specified in the instructions supplied with the rings; it is permissible for it to be greater than specified up to a maximum of approximately .030-inch. Install the rings on the correct pistons so they will be used in the cylinders in which they were checked.

Assemble the crankshaft and its associated parts in the cylinder block, using the same precautions observed when the crankshaft bearing clearances were checked. Be sure that main bearing inserts are installed in the bores they were in when their clearances were checked and that connecting rod bearings inserts are installed in the rods in which



After the rod and piston weights have been equalized, crankshaft's balance is checked and corrected on a special machine.

they were checked. Torque all capscrews and bolts to their specified tensions. Install the camshaft by carefully inserting it in its bearings so its lobes won't scratch the bearings. When installing the camshaft's drive sprockets and chain be sure the chain is installed so that the timing marks on the sprockets will be in their correct relationship to each other. Correct valve timing is essential to good engine operation.

Work on the cylinder heads should be begun by running a reamer of the correct diameter through their valve guides. Special valve guide reamers are available from some auto parts stores and tool supply houses. It's doubtful whether guides with bores smaller than they should be will be found in late-model engines but only a few minutes are required to be sure.

The valve seats should be reground with a good seat grinder. The larger diameter of the new seats should be just a few thousandths of an inch smaller than the larger diameter of the valve faces. Intake seats should be approximately 1/16-inch wide and exhausts should be approximately 3/32-inch wide. After establishing the outer diameters of the seats, grind their inner circumferences and the upper ends of the valve ports with a 60-degree stone. This will blend the seats into the ports. Regrind the faces on the valves to the correct angle with an accurate refacing machine.

After the valve seats have been reground and the valves refaced, the volumes of the combustion chambers, in heads that have them, must be measured. This is done by supporting the head with its gasket surface up and level and then measuring the quantity of water required to fill each of the chambers level with the gasket surface. The valves and spark plugs must, of course, be in the heads when this is done. Springs aren't necessary on the valves.

Care must be taken to not overfill the chambers. This is possible because water's surface tension allows it to rise a fraction of an inch above the surface around the chambers before it flows onto the gasket surface.

The best method of measuring combustion chamber volumes is by making a special plate from a piece of 3/8-inch or 1/2-inch thick clear Plexiglas that can be bolted to a head's gasket surface. A hole approximately 1/4-inch in diameter drilled through the plate near the edge of each chamber is used as a filler opening for the liquid and to let air in the chambers escape. Such a plate is easy to make once the Plexiglas has been obtained. It will simplify the measuring job considerably and the results will be much more accurate.

Combustion chamber volume is important because it has an influence on compression ratio. Volumes greater than they should be will lower the ratio and smaller volumes will raise it. For the compression ratio to be as high as possible and the engine still legal, the volumes should be the minimum allowed by the factory tolerance. The only way to reduce volumes that are greater than they should be is by removing material from the head's gasket surface by milling or grinding. This must be done carefully to prevent reducing the volumes too much.

Each of the valve springs should be checked individually for free length and pressure. Springs shorter than the specification for their free length should be replaced. Those that don't exert a pressure equal to that of the maximum for their tolerance when they are compressed in a spring testing machine must also be replaced. When replacing springs for any reason be sure to test the new ones before using them. It isn't at all uncommon for new springs to be defective in some way. Springs that aren't up to specifications for free length and pressure can't exert the specified pressures on the valve stems.

When assembling the heads be sure to adjust the valve springs' installed length. This is the length of the springs when they are locked to their valve stems with the retainer washers. An installed length longer than specified will reduce the pressure the springs exert on the valves. A length shorter than specified will increase the pressure. Low pressure can allow the valves to float at a lower rpm than they should and high pressures can increase tappet and cam lobe wear.

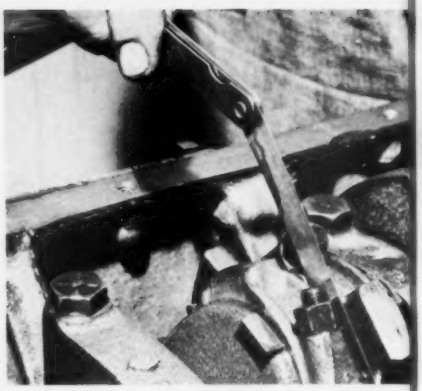
One way to measure spring installed length is by installing a retainer washer and its split locks on the valve and then exerting pressure on the washer with the fingers to hold the valve tightly against its seat while the distance between the spring seat on the head and the seat on the washer is measured with a steel rule. If the measurement is greater than the specified installed spring length it can be reduced as necessary by placing the required number of special shim washers on the head. When making the measurement be sure to allow for oil shields of the stamped steel type that may be inserted between the retainer washer and the spring's upper end. It's not probable that the measurement will be less than specified but if it is the best correction would be to try different valves. The measurement could be increased by grinding the valve faces and the seats in the head but neither of these would be desirable because they sink the valves deeper into the heads. This could affect flow around the valves and also increase the volumes of the combustion chambers.

Installing cylinder heads on the cylinder block is merely a matter of cleaning all the gasket surfaces and both sides of the gaskets. There must not be any foreign matter on the surfaces that might prevent the members from bolting together as they should. Usually, a better seal can be obtained between the gasket and the head and block by painting both sides of the gasket, or the surfaces of the head and block, with aluminum paint. Paint used for this purpose should be the type recommended for hot surfaces. Allow it to dry for a few minutes and then install the gasket and the head.

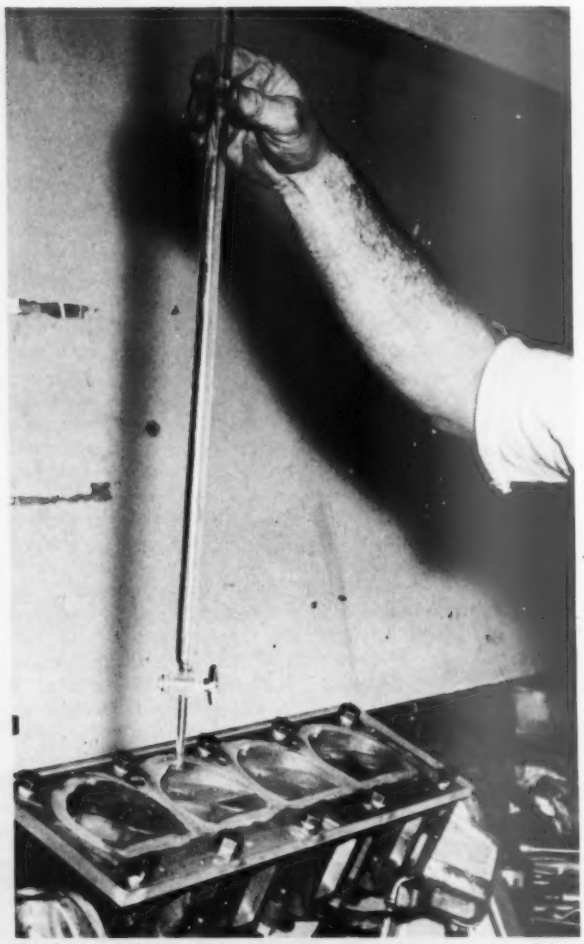
Threads in some cylinder blocks for cylinder head capscrews extend into the block's water jacket. Threads of

This 409 Chevy connecting rod and piston assembly, designed for a pressed-in pin, has been reworked to take a floating pin.

To get the highest legal compression ratio in a 409 Chevrolet, the depth of the valve relief on the piston heads must be correct.



Endplay of the connecting rods on their crankpins must be checked. This can be done with the blade of a thickness gauge.



Combustion chambers being measured with a liquid to determine their volumes and any differences existing between the volumes. The Plexiglas plate makes the job accurate and easy.

capscrews used in these blocks should be coated with a good sealant that will prevent water's seeping past the threads and then up past the capscrews. This water could, possibly, seep into the rocker arm chamber and be carried into the crankcase with the lubricating oil that drips off the rocker arms and valve assemblies.

It's important that the capscrews be tightened carefully to their specified torque. For most engines there is a definite tightening pattern but a pattern that can be used successfully on any engine consists of starting in the middle of the head and working outward toward the ends. Start with the middle row of capscrews, then tighten the next row on one side of the middle, then tighten the row on the other side of the middle, then back to the first side, and so on until all the capscrews have been tightened.

Don't tighten the capscrews completely the first time over. Run them all down against the head with a speed wrench and then go over them with a torque wrench adjusted to a tension thirty or so pounds below the specified tension. The next time over increase the tension ten pounds or so. Continue in this manner until each capscrew has been tightened in three or four steps and the specified tension has been reached. Two or three times over the pattern at the specified tension will guarantee that all the capscrews are pulled down securely.

Installation of pushrods and rocker arm assemblies doesn't

Correct valve spring pressure is extremely important to efficient valve action at high engine speeds. Here, a spring's pressure is being measured at its specified installed length.

Deck clearance is measured with a depth micrometer when the piston is at top center in its cylinder. It must be within the tolerance.

FAR RIGHT—Valve spring installed length being adjusted with the same gauge used for determining its length when its pressure was measured.

require anything more than common sense, and adjusting the valve lash shouldn't be any problem. The main concern with the lash is to be sure it isn't less than specified for any of the valves. After the engine has been started and brought up to its normal operating temperature the lash must be readjusted while the engine is hot. This adjustment must be made accurately.

Correct installation of the rest of the engine's parts requires nothing more than just good mechanics. If the cylinder block or heads, or both, were machined to adjust the compression ratio it may be necessary to elongate the capscrew holes in the intake manifold. The manifold's gasket must be aligned carefully on the heads so they don't overlap the heads' ports. Any overlapping of this nature will form a restriction in the port that will reduce the flow of fresh fuel and air mixture from the manifold to the heads' passages. Ports in the manifold must be aligned as well as possible with those in the heads. Sometimes accurate alignment isn't possible without grinding either the head ports or the manifold ports but grinding of this nature usually isn't allowed by drag associations. As a rule manifold ports are made slightly smaller than head ports to reduce the possibility of any overlapping.

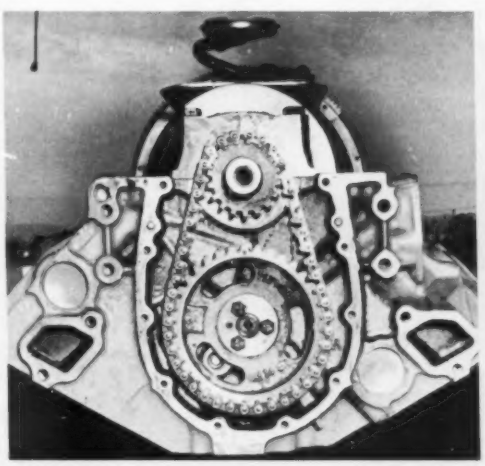
The carburetor, or carburetors, should be examined to determine whether their throttle valves open fully. Any adjustment to their linkage to make them reach their full-open position should be made. Another check that must be made after the engine is in its chassis and the car's accelerator pedal linkage has been connected to the carburetor linkage is to determine whether movement of the accelerator pedal opens the valves fully. If it doesn't, adjust the linkage so it does.

Gaskets between carburetors and the intake manifold must not overlap the bores in either the manifold or the carburetors. Also, the carburetor bores must not overlap those in the manifold. It's doubtful whether any of these conditions will exist but the possibility should be checked. Any overlapping condition should be corrected by making the necessary alterations to the parts involved.

The ignition distributor's breaker point adjustment and the advance curve provided by its advance mechanism should be checked on a distributor testing machine. For best results the number of degrees the governor weights advance the timing at each of the test speeds should be the maximum allowed by the distributor's specifications.

Spark plug cables that have either copper or stainless steel conductors should be used between the distributor and the engine's plugs. If the engine had TVRS nonmetallic conductor cables as standard equipment replace them with a new set that has metal conductors. TVRS cables are good enough for street engines as long as they are in good condition but because of their built-in resistance it's not advisable to use them on a high-performance engine.

DON FRANCISCO



Correct timing chain installation on a 409 Chevy is simply a matter of aligning the timing marks on the sprockets with the shaft centerlines.

FAR RIGHT—One of the best preventatives for head gasket blowing is to tighten the cylinder head capscrews with an accurate torque wrench.

REWORKING A SUPER STOCK

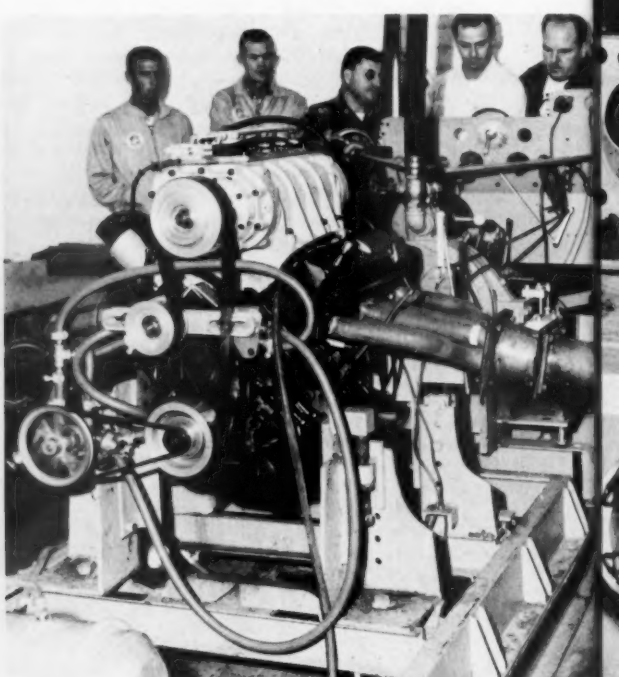
THEIR large standard displacement and rugged construction make the new Super Stocks good bases for modifications to convert them into all-out competition engines. This has been proved by several red-hot Pontiac's and a few Chrysler's. Less is known at this time about the 390 Ford and 409 Chevy because they are fairly new but it won't be long before more rodders are giving them the treatment.

The only thing against the Super Stocks for all-out conversions is their weight. Compared to the little Chevy V8's, which make excellent competition engines, they are quite heavy. Engine weight is of minor importance in cars to be used for top-speed straightaway runs but for any kind of track racing it can be a handicap in any car except one that has a stock body. It is extremely important in drag racing cars but despite this the tremendous torque and power outputs these engines are capable of developing give them the edge over most lighter engines.

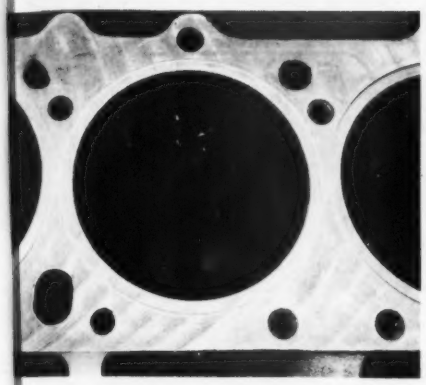
Procedures for modifying standard passenger car engines for all-out competition have become so standardized that all a fellow has to do is list them and then make up his mind just how far he wants to go with each. Listed in their logical order the procedures are:

Boring
Stroking
Raising the compression ratio

This Pontiac, on the dynamometer at Edelbrock Equipment Co., in Los Angeles, is fitted with log manifold and six two-throats.



Getting the most out of a reworked engine requires equipment like this engine dynamometer at Ed Iskenderian Racing Cams.

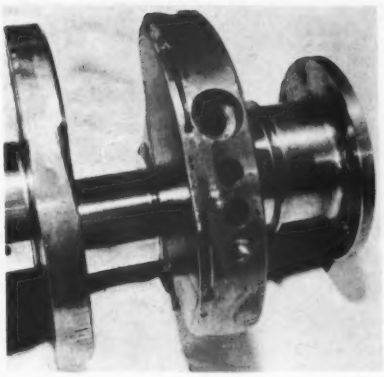


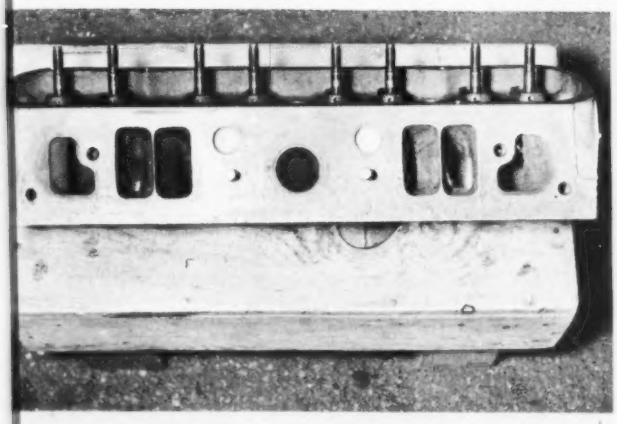
LEFT—Grooves of type shown here are often cut around cylinders of engines to be reworked for rings made from copper wire that prevent gasket blowing.

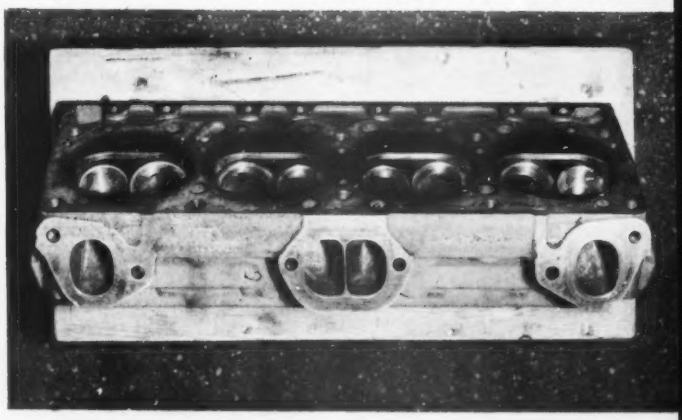
RIGHT—The steel plate welded to end counterweight on this stroked crankshaft makes it possible to balance the shaft.

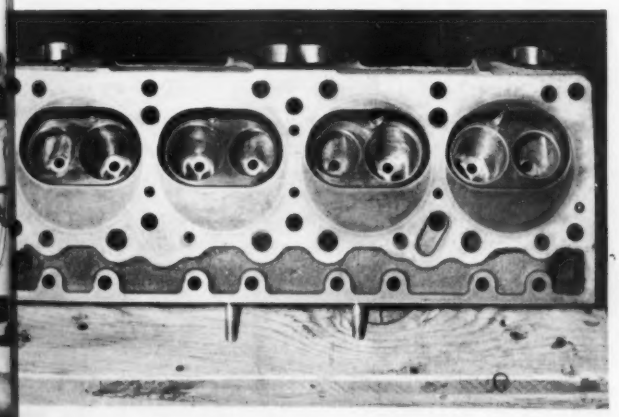
BELOW LEFT—Porting is an important part of any engine reworking job. Intake ports in this Pontiac head were enlarged.

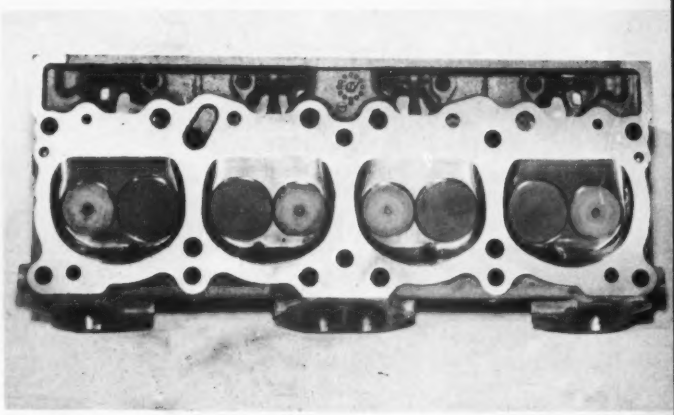
BELOW RIGHT—Getting exhaust out is as important as getting fuel and air in. These are enlarged Pontiac exhaust ports.











ABOVE LEFT—Valve ports in this head were enlarged for oversize valves to reduce the restrictions valve heads cause.

ABOVE RIGHT—Pontiac head was ported and fitted with oversize valves and shape of its combustion chambers changed.

RIGHT—One of forged aluminum pistons being manufactured by Mickey Thompson. Forged pistons are made for most engines.

LEFT—This is a pair of the forgings from which Mickey's pistons are machined. The ribs add strength to the pistons' skirts.

LEFT—A forged aluminum piston on a forged aluminum rod makes a sturdy assembly that will take a terrific beating.

BELOW—Some engine builders prefer boxed connecting rods such as this one, with its piston, for a big Chrysler "B" engine.

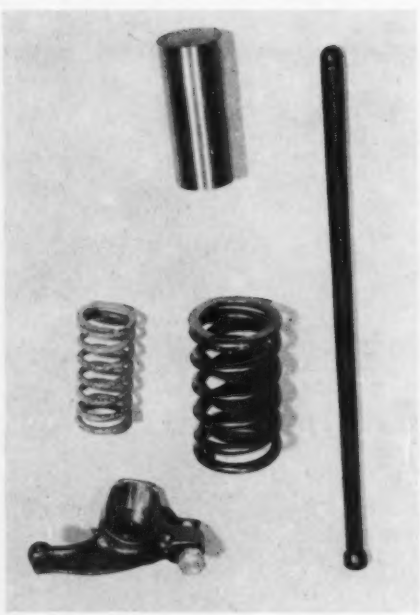
Balancing
Porting
Installing oversize valves
Installing a reground camshaft
Installing additional carburetion or a blower
Installing a special ignition distributor or magneto
Installing headers.

As the reason for reworking an engine for all-out competition is to get as much performance as possible out of it, each modification should be exploited to the practical maximum. This is where the Super Stocks regain their individual identities because there are limits to the amounts each of them can be modified by some of the procedures. This applies especially to boring and stroking.

The maximum amount the cylinders in any engine can be bored oversize depends entirely on the original thickness of their walls. The walls, after boring, must be thick enough to have adequate strength to withstand the high normal combustion pressures created on power strokes and the even higher pressures that result from detonation.

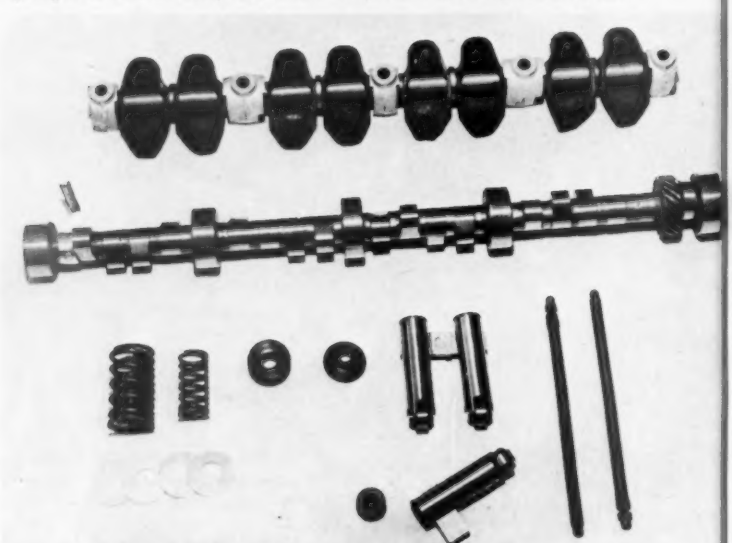
For most late-model engines experience has shown the boring maximum to be ⅛-inch. This is for naturally-aspirated engines. However, the ⅛-inch oversize rule shouldn't be accepted at face value because there are some cylinder blocks that cannot be safely bored this amount. For blown engines it is sometimes recommended that cylinders be left at their standard bore diameter so they will have the maximum possible strength.

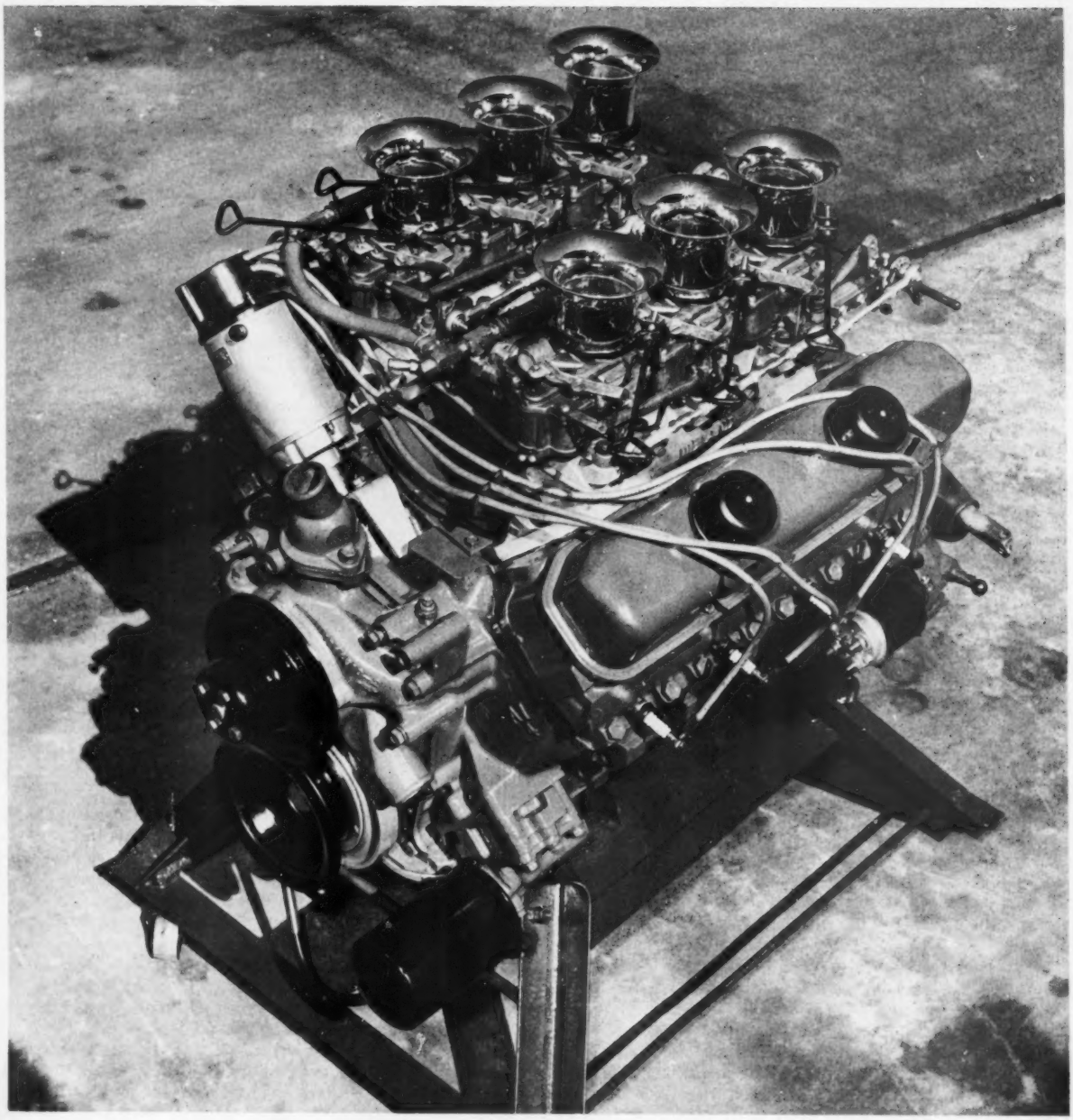
Little is known at the present time about boring Chevy 409 blocks. Cylinder walls in 409's are very thick except in the areas where the cylinders are adjacent to each other. By applying the known factors involved, which are the distance between the centers of the cylinders, the original bore diameter, and the fact that there is clearance between the cylinders, a simple mathematical computation shows



Special valve actuating parts for a Chrysler "B" engine include a lightweight adjustable rocker arm, dual valve springs, solid valve lifters, and stiff but light tubular pushrod.

With the stock pressed-steel rocker arms for a Chrysler are a roller tappet camshaft, roller tappets, dual valve springs, hardened-steel spring washers, and adjustable tubular pushrods, all by Iskenderian.





This is one of the three Chrysler Corporation engines, originally of 413 cubic inches, Norm Thatcher built for his Bonneville Dart.

the standard cylinders to be approximately .2012-inch thick where they are closest to each other. This doesn't leave much extra material for boring to a large oversize.

Norm Thatcher, the Chrysler expert in Van Nuys, Calif., says that Chrysler 413 blocks shouldn't be bored at all if the engine is to be fitted with a blower, and that the maximum for naturally-aspirated engines should be 1/16-inch. Fellows have bored these blocks as much as 1/8-inch and used them but the odds against this are not good. The problem is that standard walls may not have a uniform thickness around their circumference because the cores that determine the locations of the cylinders in the cylinder block during the casting process aren't always positioned accurately in the mold, and sometimes the cores slip during the casting.

Les Ritchey, the tuner with the magic touch at Performance Associates in Covina, Calif., and one of Southern California's principal Ford champions, recommends 4 1/16 inches as the maximum diameter for bored cylinders in Ford 390 engines. This is .080-inch larger than standard cylinders. Here, again, the problem is one of original wall thickness as determined by core problems.

Mickey Thompson, who has probably built more all-out Pontiac engines than anyone in the country, recommends that cylinders in V8 blocks be bored a maximum of .060-inch for blower installations and 1/8-inch for naturally-aspirated engines. He says many blocks have been bored larger

than these recommendations and used but it's his experience that larger bores aren't capable of withstanding the really high combustion pressures created in an engine that is running right, such as those built in his shop. Cylinder walls that flex are the cause of power losses and they also allow excessive quantities of lubricating oil to enter an engine's combustion chambers. This upsets fuel and air mixtures and causes spark plug problems that can result in further power losses.

Cylinders bored to large oversizes require special pistons because factory replacement pistons are seldom made in sizes larger than .060-inch oversize. This isn't any problem because pistons strong enough for any application are readily available from several manufacturers for engines of all the popular makes. Pistons of this type are available in both cast aluminum and forged aluminum types.

The welded crankshafts that are the standard of the stroker shaft business have made crankshaft stroking a highly successful means of increasing the displacement of all popular engines. Engine design factors that limit the stroke increase that can be used in an engine are the amount of material that can be removed from the crankshaft's crankpins to relocate the pins' centers for the new stroke, the clearance in the engine's crankcase, and the compression height of the pistons.

As of this time no one has stroked a 409 but there isn't any apparent reason why these engines can't accommodate a stroke as long as that that can be used in 348, which is 3⅞ inches.

Norm Thatcher has stroked Chrysler 413 engines a full ¾-inch but he considers this to be the maximum. The limiting factor is the depth of the notches that must be ground in the lower ends of the cylinders to provide clearance for the connecting rod nuts. For a ¾-inch stroker these grooves have to be quite deep. Norm is afraid that going much deeper would put a fellow into the block's water jacket. Notches for a ½-inch stroker aren't too bad, and just knocking the fuzz off the block provides ample clearance for a ¾-inch stroker.

Les Ritchey recommends a maximum stroke increase of ½-inch in 390 Ford's. He says there is ample clearance in the engine for a ¾-inch stroke but he feels that a stroke longer than ¾-inch upsets the engine's bore to stroke ratio too much to be practical.

Mickey Thompson strokes Pontiac V8 crankshafts to 4⅜ inches, which is ¾-inch longer than stock, without any major problems. For the longer strokes, notches must be ground in the outer edges of the crankcase for connecting rod clearance but this isn't any problem.

Stroking doesn't present any problems as far as connecting rods and bearings are concerned because the crankpins on all welded stroker shafts are built up to their standard diameter. This makes it possible to use standard bearing inserts and connecting rods on them. For an engine that is to have a blower or run on highly nitrated fuels, it's recommended that standard connecting rods be replaced with special aluminum rods. These are expensive but worth their cost in an engine that will possibly deliver twice its rated power output and run at crankshaft speeds considerably higher than for which it was designed.

The correct compression ratio is very important in all-out modified engines. The best ratio for a specific engine will depend on whether the engine is to be blown or naturally aspirated, and whether it is to run on highly nitrated fuels, straight alcohol, or gasoline.

The reason for establishing the desired compression ratio at this point in the modification procedure is that in most engines it is adjusted with the pistons. The compression ratio for any engine is determined by the compression height of its pistons, the shape of the pistons' heads, the volume of the combustion chambers in its cylinder heads, and the individual displacement of its cylinders. Cylinder displacement is fixed for any bore and stroke combination and combustion chamber volume is fixed for a specific type of heads. As piston design is fairly flexible, and new pistons will be needed anyway in a bored or stroked engine, it becomes logical to regulate the compression ratio with the pistons. Altitude, tuning factors, and valve timing have definite influences on the highest usable compression ratio for any engine.

Compression ratios ranging from 7.50 to 1 to 10.00 to 1 have been used in blown engines. The higher ratios are often satisfactory in engines used for drag racing where the throttle is open for only a few seconds but it's doubtful whether they would be suitable for longer full-throttle runs where the combustion chambers would have time to become overheated.

Compression ratios for highly nitrated fuels have approximately the same range as those for blown engines. Here, again, a ratio that might be too high for long runs might be satisfactory for drag racing.

For straight methanol, compression ratios as high as 13.00 to 1 or 14.00 to 1 can be used without trouble. Alcohol's high resistance to detonation and the rich mixtures it requires give it the ability to withstand high ratios. The only problem is that in some engines obtaining a ratio as high as desired may be difficult because of combustion chamber configuration.

A ratio of 11.00 to 1 is high enough in any engine that will run on gasoline. For satisfactory results the combustion chambers will have to be kept free of carbon deposits and care taken to keep fuel and air mixtures delivered by the engine's carburetion system on the rich side. Lean fuel mixtures and high compression ratios don't go well together. Also, the best available gasoline will be required.

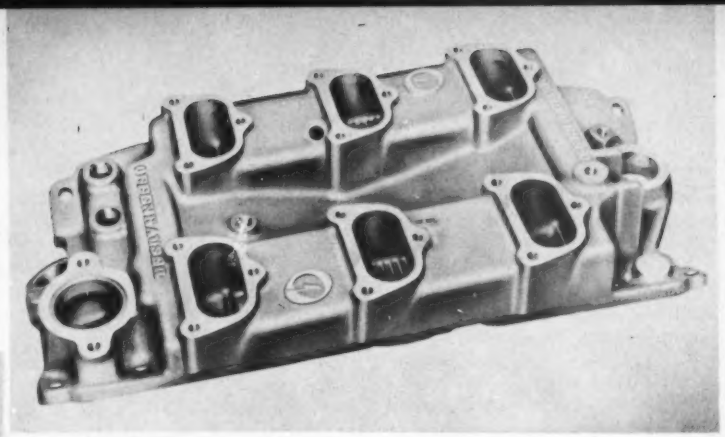
After any boring or stroking job, or any other operation that changes the weights of an engine's connecting rods or pistons, the crankshaft and the rods and pistons must be balanced as an assembly. This is a job that can't be avoided. It is as necessary as the parts themselves. An engine will run if it isn't in balance but the length of time it will run without literally shaking itself to pieces is limited. Also, correct balance is as important to maximum crankshaft rpm's as it is to smooth running and long engine life.

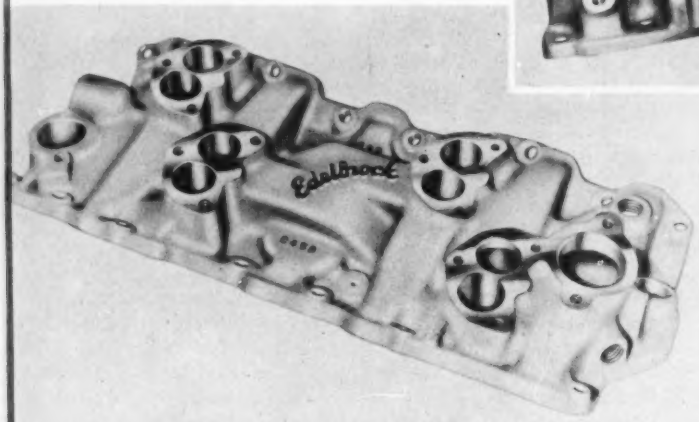
Intake and exhaust ports and passages in all the Super Stock engines are quite large in their stock form but for an all-out engine it would be advisable to enlarge them as much as practicable by grinding their surfaces. This is a job some rodders could possibly do but it requires small grinding stones and a die grinder or high-speed flexible shaft to rotate the stones that the average rodder doesn't have. Also, the job is a tedious one and it's possible for an inexperienced fellow to grind through a passage's walls and into the cylinder head's water jacket. Sometimes these goofs can be repaired by plugging them with brazing or silver soldering methods but a set of heads are considerably better if they don't have such repaired spots. When all the angles are considered a fellow is better off if he can have the porting done by someone who specializes in this sort of work. Porting jobs aren't inexpensive but they can be the source of extra horsepower.

One of the main reasons for porting is to make all the intake or exhaust ports and passages in the heads the same shape and of the same cross-sectional areas. Valve port ends of the passages should be blended into the valve ports as smoothly as possible.

Porting is especially important if oversize valves are installed in the heads. The purpose of oversize valves is to

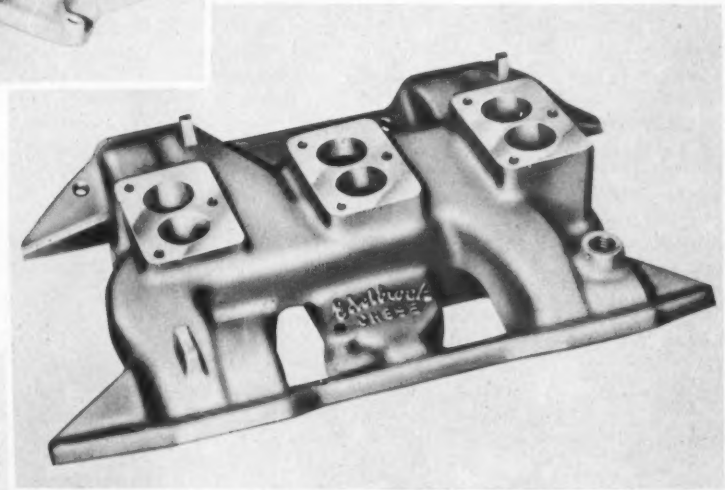
This Offenhauser six-carburetor log manifold is made in two types for 265, 283, 348, and 409 cubic inch Chevy V8's. It uses two-throat, three-bolt flange carburetors.

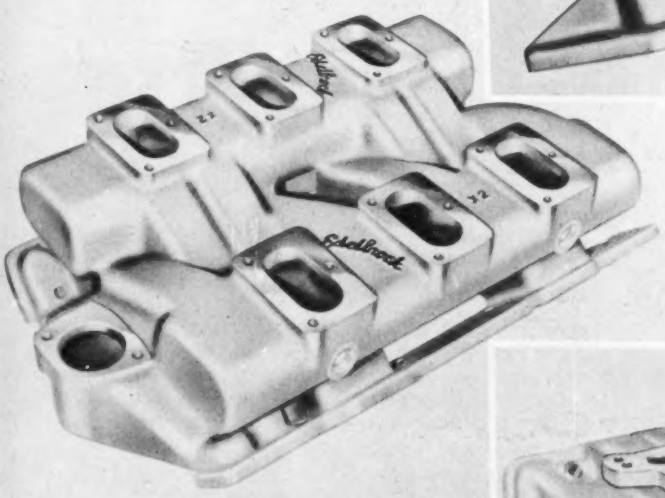




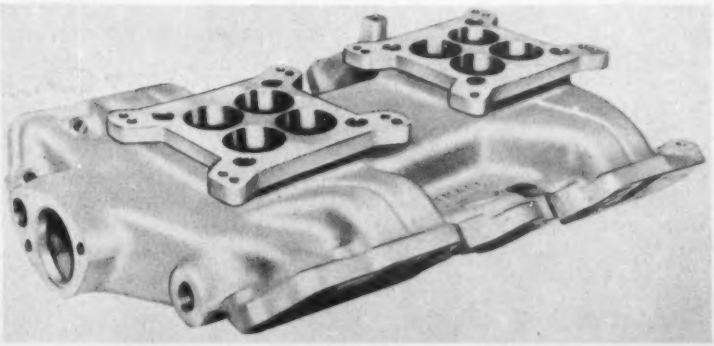
Edelbrock's "Staggered four" manifold for 348 and 409 Chevy's has its four two-throat carburetors arranged to make it a 180-degree type. It is exhaust-heated for good low-speed performance.

Another Edelbrock manifold is for '59 and '60 Chrysler engines. It is for three two-throat carburetors, is of the 180-degree type, and is exhaust heated.





Edelbrock's latest log manifold is the "Ram Log." It showed impressive horsepower gains during tests on Edelbrock's engine dynamometer. It is currently available for Chevy and Pontiac V8's.

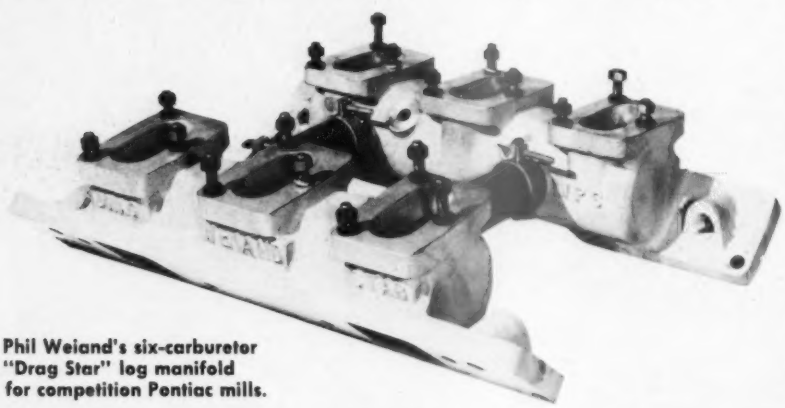


Offenhauser manifold for two four-throat carburetors is for '55 through '60 Pontiac V8 engines. It uses standard carburetors, is exhaust heated for street driving.

allow greater quantities of fresh mixture and exhaust gases to flow through the valve ports when the valves are open but this isn't of much value if the ports and passages that serve the valve ports are so small that they restrict the flow.

The heads on oversize valves used in most engines are usually ⅛-inch larger in diameter than the heads on standard valves. In the Super Stock engines that have combustion chambers in their cylinder heads, which include all but the 348 and 409 Chevy's, installation of oversize valves reduces the distance between the head of an open valve and the side of the combustion chamber. This reduced clearance can nullify much of the benefit that should be received from the oversize valves by restricting the flow of fresh mixture or exhaust gases past the valves' heads. To allow the valves to be as effective as possible the combustion chambers should be enlarged around the valve seats by removing material from their surfaces adjacent to the seats. The thing that limits the amount chambers can be enlarged in this manner is the thickness of the combustion chamber walls. The walls must not be made so thin that they can't hold combustion pressures. Material can be removed for this purpose with either small grinding stones of the type used for porting or with reamers that can be piloted in the valve guides.

Things to consider before installing oversize valves are the clearance between the heads of the standard valves for each cylinder and the weight of the oversize valves. In some engines, such as the 340 and 350 horsepower 348 Chevy's and the 409, space between the heads of standard valves is so limited that any appreciable enlarging of the valve heads would cause them to touch. For adequate clearance when the valves get hot there should be a minimum of 1/16-inch between their heads when the valves are cold.



Phil Weiand's six-carburetor "Drag Star" log manifold for competition Pontiac mills.

As valve heads become larger, the weight of the valves becomes greater. This weight increase can sometimes be eliminated or minimized by grinding material from the heads. The main precaution when doing this is to remove the material from the correct areas so the strength of the heads won't be reduced below the minimum for safe operation and the heads' contours won't become such that fresh mixture and exhaust gas flow past them will be reduced.

Valve weight is important because it has an influence on the maximum engine speed an engine can run before valve float occurs. In a high-performance engine high crankshaft speeds are absolutely essential to good performance. After weighing these considerations and restrictions one against the other it probably will be concluded that standard valves that have been carefully lightened and recontoured for maximum flow will be adequate in a reworked 1961 Super Stock.

The importance of valve action to engine performance is a subject with which every rodder is familiar. The hotter optional cam grinds some manufacturers have available for their Super Stock engines do very well in all-out modified engines but in some instances performance can be improved by installing a shaft obtained from one of the many companies that specialize in grinding high-performance shafts. One limitation of factory optional camshafts is that they are designed for engines that have their standard displacement. More than likely they won't provide enough valve opening area, as determined by the volumes of fresh mixture and exhaust gases that can actually flow past the valve heads rather than by the number of degrees of crankshaft rotation the valves are off their seats, for an engine in which the displacement has been increased by boring and stroking.

Companies that specialize in camshaft grinding have grinds for engines of all sizes. Finding the one that will do the job in a specific engine isn't a problem of availability but of choosing between the many that are available. As there isn't any formula a fellow can apply to the camshaft selection problem, all he can do is base his decision on what he has seen and heard and on the recommendations made by the men who grind the shafts. One rule a fellow absolutely should not break when buying a special camshaft of any type is to discuss his requirement with the man whose product he has chosen to use. These men know the effect their camshafts will have in an engine. Letting them know the type of engine you are building and the way it will be used can save you time and money. The wrong cam grind in any engine can be a discouraging thing.

Something as important as selecting the camshaft you think will do the best job is installing the correct related parts with the one you buy. The company that ground the shaft will simplify this for you by having available an installation kit that includes the special parts they recommend for your engine. Included in the kit could be valve lifters, pushrods, rocker arms, valve springs, and valve spring retainer washers. Parts in these kits have been designed to function as an assembly. They let the camshaft do the job for which it was designed and, in addition, lengthen the service life of its cams and the related parts to the design maximum.

The best carburetion system for an all-out modified engine is a fuel injector of the Hilborn type. But, unfortunately, at the present time these are available only for Pontiac's and 413 Chrysler's. For the fellow who has a 390 Ford or a 348 or 409 Chevy, or the rodder who doesn't want an injector for some reason, the only alternative is a multiple-carburetor setup.

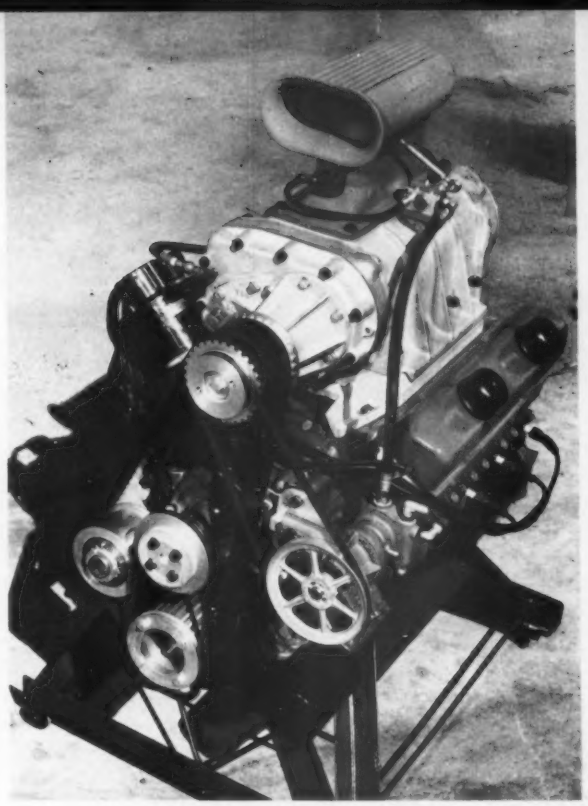
Nearly all the large-volume carburetor setups for late-model engines utilize log manifolds. These are the simplest of all manifolds for automotive-type engines. For V8's they consist of two passages, one for each cylinder bank, that have individual branches for each of the intake ports in their respective cylinder head. The logs are usually connected with comparatively small-diameter passages at one or more points that help equalize the vacuum pulsations in them. Flanges are provided on the upper sides of the logs for carburetors. Two-throat carburetors of different makes are usually recommended. The number of carburetors will vary between a maximum of eight and a minimum of four.

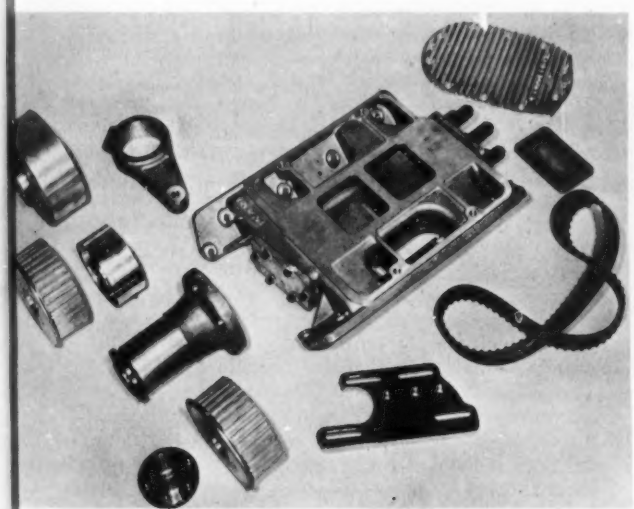
The purpose of a multiple-carburetor setup is to provide enough venturi area to let an engine's cylinders induct as much air as they can and to guarantee ample fuel flow capacity so that the percentage of fuel mixed with the air as it passes through the carburetors will be correct for maximum power output. It is difficult, and sometimes impossible, to obtain adequate flow of alcohol and alcohol-base fuels through carburetors because of the large quantities of these fuels required in relation to air flow. Extensive modifications must be made to the carburetors' float needles and seats and metering jets to increase their flow capacities to the maximum.

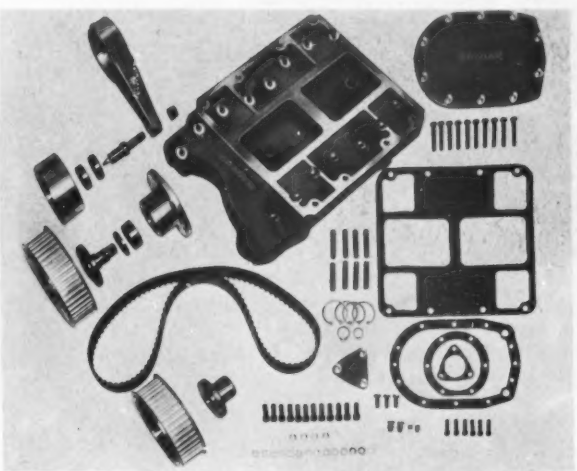
An important feature of many competition engines these days is a GMC blower. Many companies make adaptor equipment that simplifies installation of these blowers on late-model engines.

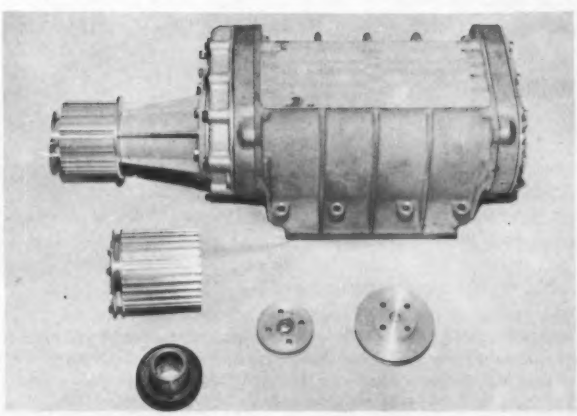
The purpose of all the modifications done to an engine to increase its torque and horsepower outputs is to enable the engine to efficiently consume a greater weight of fuel and air in a given time. A blower does this by literally forcing air and fuel mixture into the engine's cylinders. Actually, a blower is nothing but a compressor that is driven by the engine. The pressure it creates in an engine's induction system is "boost" pressure. In a naturally-aspirated engine manifold pressure is always below atmospheric but in a blown engine it can be several pounds above atmospheric.

It's possible to use carburetors on a blower but the same air and fuel flow problems that affect multiple-carburetor









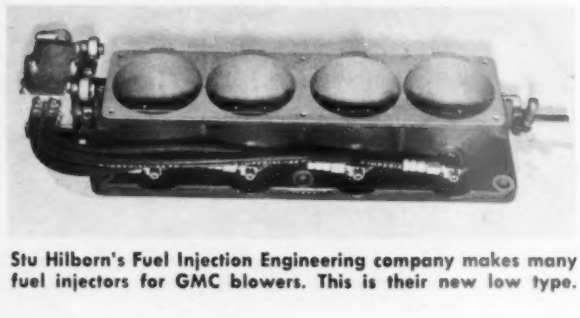
ABOVE—This is a complete GMC blower adaptor setup for a Pontiac V8. It is manufactured by Mickey Thompson Enterprises. All a fellow needs with a kit of this type is a blower.

TOP RIGHT—Blowers have come into their own for all-out competition engines. This blown Chrysler is another of Norm Thatcher's engines for the '61 Bonneville Nationals.

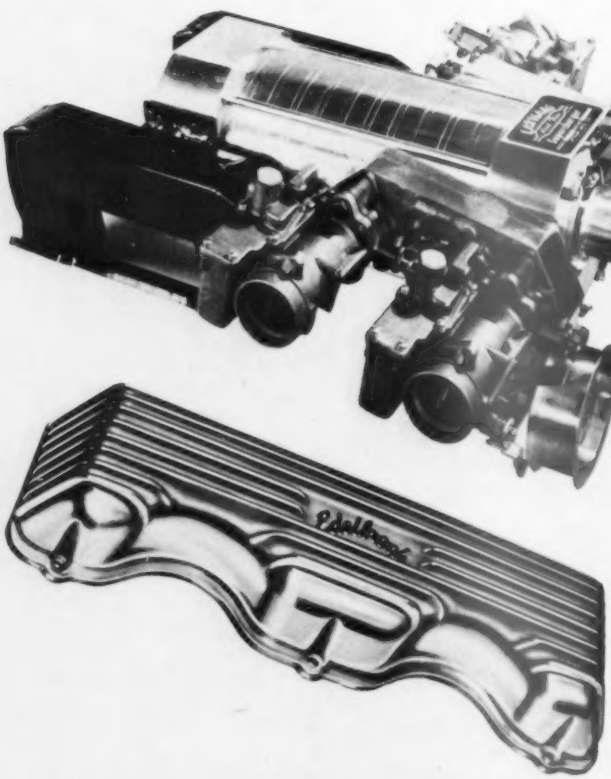
ABOVE RIGHT—Bell Auto Parts "Cragar" adaptor kit for GMC blower. Kits of this type, from different manufacturers, are available for most of the engines suitable for reworking.

RIGHT—GMC blowers are the most popular for competition engines. This is a 6-71 with timing belt pulleys and other adaptor equipment built by Tom Beatty.

This is one of the fuel injectors Enderle Fuel Injection makes for GMC blowers. An injector is almost a necessity on a GMC.



Stu Hilborn's Fuel Injection Engineering company makes many fuel injectors for GMC blowers. This is their new low type.



The blower assembly above is the Latham axial flow unit, which is made for both competition and normal driving. It is driven by a single flat belt. Below is one of the many aluminum rocker arm covers, this one by Edelbrock, available for most V8's. An engine should look good as well as go.

setups on a naturally-aspirated engine also apply to carburetor setups on blowers. A far more satisfactory arrangement is to use one of the Hilborn or Enderle fuel injectors made specifically for GMC blowers. Air and fuel flow capacities of these injectors are adequate for any engine.

All the work a rodder goes to to build a big, deep-breathing engine can be wasted if he doesn't provide adequate means to fire the compressed fuel and air mixture in its cylinders. It is impossible for a stock ignition system to overcome the high compression pressures created in either a naturally-aspirated or blown competition engine and to function at the cycling frequencies required for the high crankshaft speeds normal to competition engines. Only the best available ignition systems are equal to the job. These are limited to one or two two-coil distributor and coil combinations or a magneto of some type.

One of the best of the two-coil distributors now available is the Spalding Flamethrower. This unit is used by a few owners of highly successful competition cars. Its only disadvantage, which it shares with all battery ignition systems, is that it requires a battery. Many rodders don't like to burden their car with a battery nor do they care to spend the time necessary to keep a battery in a fully-charged condition at all times. They avoid these problems by installing a magneto on their engine.

Of the different makes of magnetos available now for competition engines the Scintilla Vertex is by far the most popular. The Vertex is a Swiss-built unit that is adapted to engines of different makes by fitting it with a base and driveshaft assembly of the correct type. This adapting is done by the companies that import and sell the mags, not by the customer. A fully-adjustable governor weight automatic advance mechanism in the mag allows an advance curve suitable for any type of engine, either naturally aspirated or blown, to be provided. The initial cost of a Vertex is high compared to that of a battery ignition system but upkeep costs are low and resale value is excellent. Also, a Vertex for one make of engine can be converted for only a few dollars for an engine of another make that has the same number of cylinders.

After thoroughly reworking an engine to allow it to consume as much air and fuel as it possibly can, care must be taken to complete the circuit the air that enters the carburetor or fuel injector must follow to again reach the atmosphere. This means that the engine's exhaust system must be as free-flowing as its induction system.

Exhaust systems for competition engines are often regulated by competition rules. Sometimes the exhaust must be discharged at a certain location in relation to the car's wheels or cockpit, or the pipes must not be aimed in a certain direction, etc. When such restrictions apply, the only thing a fellow can do is design the system as best as possible within the limitations.

The two basic types of exhaust systems are those that have headers of some sort and those that have an individual pipe for each of the engine's exhaust ports. Both types have been used satisfactorily on competition engines but when the decision is left entirely to the car owner most fellows use an individual pipe for each port. This is usually the end of similarity between the systems because choices of pipe diameters and lengths vary radically.

Because of the many factors involved in the choice of an exhaust system, this is another of those modifications for which a fellow must use his experience to determine a starting point and then not be afraid to make changes in an effort to find the best system for his particular engine and car.

DON FRANCISCO

MATH and FORMULAS FOR HOT RODDERS

BEING able to make mathematical computations that involve many of the specifications that pertain to his car's engine and chassis and to the car's performance will help a fellow become a successful hot rodder. A college degree in mathematics isn't necessary to be able to make these computations. All that are needed are the necessary formulas and a basic understanding of grammar school arithmetic.

A mathematical formula is nothing more than a pattern for solving a specific problem. It places the various factors involved in the problem in their correct order in relation to each other so the influence the factors have on each other can be computed.

The first step in using a formula is to insert the numerical values of the factors involved in their correct positions in the formula. This changes the formula to an "equation." The equation is used for the mathematical process of solving the problem.

Equations for rodding formulas are not complicated. However, it is essential that the various mathematical symbols used in them be understood so that the mathematical processes necessary for their solution can be done properly and in the correct order. The symbols indicate simple division, multiplication, subtraction, and addition.

The symbol for division is a straight line that separates two numbers placed one above the other. The lower number is always divided into the upper number:

$$\frac{12}{3} = 4$$

The symbol for multiplication is "×." It is used to separate two or more numbers in a row. For example:

$$3 \times 4 = 12$$
$$3 \times 4 \times 8 = 96$$

Numbers to be multiplied together may be multiplied in any order. The result will be the same regardless of the order used.

The symbol for subtraction is the standard minus sign. This is nothing more than a dash. It separates two or more numbers.

The number on the right of the symbol is always subtracted from the number on the left of the symbol. For example:

$$5 - 3 = 2$$
$$25 - 10 - 6 = 9$$

When more than two figures are separated by subtraction symbols the subtraction must be carried out from the left to right if the result is to be correct. For example, for the problem 25 − 10 −6, 10 from 25 equals 15, then 6 from 15 equals 9.

Addition is indicated by the + symbol. The symbol is used to separate two or more numbers. For example:

$$3 + 4 = 7$$
$$3 + 4 + 8 = 15$$

Numbers separated by addition symbols may be added together in any order.

When solving an equation that involves division as well as other steps, do all the division steps first to reduce those parts of the equation to their numerical value. Multiplication, subtraction, and addition can then be accomplished as they appear in the equation by starting at the left end of the equation and working toward the right. Completing the division first also includes those division parts that require multiplication, subtraction, or addition steps:

$$\frac{8 \times 6}{12}$$

This would be reduced by multiplying 8 times 6 and then dividing the product by 12. This part of the equation would then become 4.

When any of the factors in a formula are in parenthesis, solve the equation formed by these factors first and use the result as a single number in the overall equation. For example, the equation

$$2 \times (16 - 4 \times 3) =$$

is reduced by solving

$16 - 4 \times 3$, which is
$16 - 4 = 12$
$12 \times 3 = 36$

The main equation then becomes

$$2 \times 36 = 72$$

For use in formulas, fractions should be converted to their decimal equivalents. The easiest way to do this is with a conversion chart. Charts for this purpose are available from many sources. They are included in all types of mathematical handbooks and are stamped on some types of precision measuring instruments.

The various mathematical processes can be simplified by carrying the results to only two or three decimal places. Shortening the results in this manner will not have any detrimental effect on the accuracy of the final result.

Some formulas contain "constants." A constant is a number that remains the same regardless of the other numbers used in the formula and the resultant equation. It is a number without which the equation cannot be solved correctly.

Rodding formulas apply to many phases of the sport. The answers they give can often pave the way to performance increases and, quite often, are necessary for completing entry blanks for different events. When it is needed, one formula is as important as another. However, some formulas are used more than others. We'll take them in the general order of their popularity.

ENGINE DISPLACEMENT

A rodder should be able to compute the displacement, or "swept volume," of his engine. Most entry blanks for competitive events require engine displacement information because of class restrictions. It is also good to be able to compute displacement so that changes in it resulting from boring and stroking can be determined.

Factors involved in the displacement formula are the bore diameter of the engine's cylinders, the length of the piston stroke, the number of cylinders in the engine, and a constant. The constant is .7854, which is one-quarter of 3.1416, another constant known as "pi." Pi is used in formulas concerned with the dimensions of circles.

Actually, the engine displacement formula is the standard formula for computing the volume of a cylinder of any type with an added factor that represents the number of cylinders in the engine. The cross-sectional area of the cylinders is determined and then the volume of the individual cylinders is computed by multiplying the area by the stroke length, which is the equivalent of the length of the cylinders. Multiplying the result by the number of cylinders in the engine gives the engine's total displacement.

The formula is:

Bore diameter × Bore diameter × .7854 × Stroke length × Number of cylinders = Displacement.

Dimensions in inches will give the displacement in cubic inches. Dimensions in centimeters will give the displacement in cubic centimeters (cc). One inch equals 2.54 centimeters; one cubic inch equals 16.38 cubic centimeters.

For example, let's consider a standard 283 cubic inch Chevy V8. These engines have a cylinder diameter of 3⅞ inches and a stroke of 3 inches. The formula, with the fractions converted to decimals, becomes 3.875 × 3.875 × .7854 × 3 × 8 = Displacement in cubic inches.

To arrive at the answer, multiply the numbers together by starting at the left of the group and working to the right. The different steps will look like this:

3.875 × 3.875 = 15.015
15.015 × .7854 = 11.792
11.792 × 3 = 35.376
35.376 × 8 = 283.008 cubic inches displacement

COMPRESSION RATIO

A cylinder's compression ratio is computed by comparing the cylinder's volume, or its displacement, with the total volume of the cylinder and its combustion chamber. Cylinder volume can be determined mathematically but combustion chamber volume must be measured with a liquid.

Cylinder volume is determined in exactly the same manner as for the displacement formula:

Bore × Bore × .7854 × Stroke = Volume of one cylinder.

To measure the volume of one of the combustion chambers in the cylinder head, install the valves and spark plug in the chamber and support the head so that its gasket surface is level. Then pour water or light oil from a graduated beaker into the chamber to fill it to its gasket surface. Do not overfill the chamber. This is possible with water and other liquids that have a high surface tension. Such liquids will rise to a considerable height above the surface around the chamber before they will flow onto the surface.

The amount of liquid poured into the chamber is determined by subtracting the quantity still in the beaker when the chamber is full from the original quantity. Most beakers are graduated in cubic centimeters (cc), making it necessary to convert the result to cubic inches. However, the displacement of the cylinder can be converted to cubic centimeters. The compression ratio arrived at with the formula will be the same regardless of whether cubic inches or cubic centimeters are used. The only precaution is that all volumes used in the formula be quoted in the same terms.

The volume of the cylinder opening in the head gasket must be computed by multiplying its area in square inches by the gasket's thickness in thousandths of an inch. Sometimes it is necessary to roughly calculate the square inch area of the opening but the calculation can usually be made with sufficient accuracy that it won't affect the final computation. The volume of the opening is added to the combustion chamber volume.

Another thing that must be taken into consideration is the volume of the area between the top of the piston and the top of the cylinder block when the piston is in top dead center position. Compute this volume by measuring the distance from the top of the block to the piston head as accurately as possible with a depth micrometer or some other precision measuring device and then multiplying the area of the cylinder by the depth. The formula for this step is:

Bore × Bore × .7854 × Distance from top of piston to top of cylinder block = Volume

This volume is added to the total volume of the combustion chamber and head gasket opening. The total of these three volumes is the "final combustion chamber volume."

After the factors just described have been computed, they are applied to the following formula:

$$\frac{\text{Final combustion chamber volume} + \text{Cylinder volume}}{\text{Final combustion chamber volume}} = \text{Compression ratio}$$

For an example let's dream up an engine that has a final combustion chamber volume of 5 cubic inches and a cylinder volume of 45 cubic inches.

Applying these figures to the formula we get the equation:

$$\frac{5 + 45}{5} = \frac{50}{5} = 10$$

The compression ratio is 10 to 1.

This method of computing compression ratio cannot be used accurately for engines that have pistons with either domed or irregularly shaped heads. Any irregularity on the piston heads will make it impossible, with normal means, to determine the final combustion chamber volume because the volume displaced by the piston heads cannot be readily computed. The only way to determine the final combustion chamber volume when such pistons are used is by measuring it with liquid while the cylinder head is bolted to the cylinder block and the piston is in top dead center position.

GEAR RATIO—SPEED RELATIONSHIPS

There are four versions of the formula that involves the relationships of car speed, engine speed, rear axle gear ratio, and rear tire size. By using the appropriate version any one of these factors can be determined for any combination of the other three.

To simplify the formulas a representative symbol is substituted for each of the factors. These are:

MPH for Car speed
RPM for Engine crankshaft speed
R for Rear axle gear ratio
W for Tire size

Tire size can be determined in several ways but the one that is the easiest and as accurate as any is by measuring the effective radius of a wheel and tire assembly. This is done by measuring the distance from the surface on which the tire is resting to the center of the rear axle shaft. A tire must be inflated to its normal hot operating pressure and the car must be loaded to its operating weight when this measurement is made. The measurement must be in inches. Any fraction of an inch involved in the measurement must be converted to a decimal equivalent to simplify the mathematics. When tire size is measured in this manner a constant of 168 is used in the formula.

To determine car speed for a given combination of engine speed, gear ratio, and tire size, the formula is:

$$\frac{\text{RPM} \times \text{W}}{\text{R} \times 168} = \text{MPH}$$

For an engine speed of 5000 rpm, a gear ratio of 4.00 to 1, and a tire radius of 13 inches, the equation would look like this:

$$\frac{5000 \times 13}{4 \times 168} = \frac{65.000}{672} = 96.7 \text{ MPH}$$

To determine engine speed for a given combination of

the other three factors the formula is:

$$\frac{\text{MPH} \times \text{R}}{\text{W}} \times 168 = \text{RPM}$$

Using the same figures as for the previous example, the equation becomes:

$$\frac{96.7 \times 4}{13} \times 168 = 5000 \text{ RPM}$$

To determine the rear axle gear ratio for a combination of the other three factors, the formula is:

$$\frac{\text{RPM} \times \text{W}}{\text{MPH} \times 168} = \text{R}$$

Using the figures from the previous examples, the equation becomes:

$$\frac{5000 \times 13}{96.7 \times 168} = 4.00 \text{ to } 1$$

To determine tire size, the formula is:

$$\frac{\text{MPH} \times \text{R} \times 168}{\text{RPM}} = \text{W}$$

With the example figures, the equation becomes:

$$\frac{96.7 \times 168}{5000} = 13$$

This particular set of formulas is invaluable for determining the correct combinations of factors for drag racing or maximum speed runs such as at the lakes or Bonneville. One thing to be remembered when using them is that the effective radius of a tire will increase as a result of centrifugal force as car speed increases. This will have a slight effect on the accuracy of the computations but not so much as to make them impractical. When dealing with high speeds a slight allowance could be made in the radius used to compensate for tire growth. Anything from a quarter of an inch to half an inch should be adequate for this purpose.

Tire slippage at high speeds is something else that will affect the results but slippage is extremely difficult to judge. It shouldn't be any problem when running on pavement but on El Mirage or Bonneville it could become troublesome.

FINAL GEAR RATIOS

Formulas for computing final gear ratios for different combinations of transmission and rear axle ratios involve simple multiplication. The formula is:

Transmission ratio × Rear axle Ratio = Final ratio.

For example, let's consider a Chevy passenger car transmission that has a low gear ratio of 2.94 to 1, a second gear ratio of 1.68 to 1, and a rear axle ratio of 3.70 to 1. The final ratio in low gear would be:

$$2.94 \times 3.70 = 10.87 \text{ to } 1$$

The final ratio in second gear would be:

$$1.68 \times 3.70 = 6.21 \text{ to } 1$$

Because the high gear ratio of all transmissions is 1 to 1, the final ratio in high gear would be the same as the rear axle ratio, or 3.70 to 1.

The same formula is used for computing the final ratio for transmissions fitted with overdrives. Many overdrives have a ratio of .72 to 1. For such an overdrive and a 3.70 rear axle ratio the equation would be:

$$.72 \times 3.70 = 2.66 \text{ to } 1$$

It may be possible that a fellow won't be able to compute final gear ratios because he doesn't know the ratios of the gears in his synchromesh transmission or in his rear axle assembly. In such instances it is a simple matter, if the numbers of teeth on the different gears are known or the gears are accessible so that their teeth can be counted, to quickly determine the ratios.

Transmission gears on which the teeth must be counted are the clutch shaft gear, which is the gear driven by the engine, the countershaft gears that mesh with the clutch shaft gear and the second and low gears on the transmission's mainshaft, and the second and low gears on the mainshaft. Rear axle gears on which the teeth must be counted are the pinion gear and the ring gear. The gears are then treated in pairs, the pairs being gears that mesh with each other. The ratio of each pair is determined by dividing the number of teeth on the gear that does the driving into the number of teeth on the gear that is driven. The formula for this is:

$$\frac{\text{Driven gear}}{\text{Drive gear}} = \text{Ratio}$$

For transmission gears, this means that the number of teeth on the clutch shaft's gear is divided into the number of teeth on the countershaft gear it drives and the number of teeth on the second and low gears on the countershaft are divided into the numbers of teeth on their respective second and low gears on the mainshaft.

To find the output ratio of the transmission for second gear, the ratio of the clutch shaft gear and the countershaft front gear and the ratio of the countershaft second gear and the mainshaft second gear are multiplied together. For the low gear output ratio the clutch shaft gear ratio and the ratio of the countershaft low gear and mainshaft low gear are multiplied together.

For example, one combination of gears used in a certain model Ford V8 transmission has 16 teeth on the clutch shaft gear, 28 teeth on the countershaft front gear, 24 teeth on the countershaft second gear, 22 teeth on the second gear on the mainshaft, 18 teeth on the countershaft low gear, and 29 teeth on the mainshaft low gear. To determine the second gear output ratio for this transmission the equation would be:

$$\frac{28}{16} \times \frac{22}{24} = 1.75 \times .91 = 1.59 \text{ to } 1$$

For low gear the equation would be:

$$\frac{28}{16} \times \frac{29}{18} = 1.75 \times 1.61 = 2.81 \text{ to } 1$$

Determining rear axle ratios is merely a matter of dividing the number of teeth on the pinion gear into the number of teeth on the ring gear.

The fomula would be:

$$\frac{\text{Teeth on ring gear}}{\text{Teeth on pinion gear}} = \text{Gear Ratio}$$

For example, the ratio of a gearset that has a 9-tooth pinion and a 37-tooth ring gear would be:

$$\frac{37}{9} = 4.11 \text{ to } 1$$

CAR SPEED

Formulas for computing car speeds are based on time and distance measurements. The time is measured in seconds and fractions of seconds and the distance is measured in miles and fractions of miles. The formulas are based on the fact that there are 3600 seconds in an hour. Results are in miles per hour. The formula is a constant over the number of seconds required to travel a certain distance. Value of the constant depends on the distance traveled. For a mile it is 3600. For distances longer or shorter than a mile it is the distance, in miles or fractions of miles, times 3600. For a distance of five miles the constant would be five times 3600, or 18,000. For a quarter of a mile it would be 900.

For example, let's say that a mile was traveled in 40 seconds. Applied to the formula this would become:

$$\frac{3600}{40} = 90 \text{ miles per hour}$$

For another example let's say that a quarter-mile was traveled in 10 seconds. Applied to the formula this would be:

$$\frac{900}{10} = 90 \text{ miles per hour}$$

For a third example assume that a car traveled 5 miles in 3 minutes and 20 seconds, which is the equivalent of 200 seconds. Applied to the formula this would become:

$$\frac{18{,}000}{200} = 90 \text{ miles per hour}$$

HORSEPOWER AND TORQUE

Horsepower and torque are measures of engine performance. They are related to the extent that one cannot exist without the other. Torque is the measure of the amount of work an engine can do and horsepower is the measure of the amount of work done in a given time. The time factor for horsepower computations is the engine's crankshaft speed, as measured in revolutions per minute.

Measurements of an engine's torque output require a machine called a "dynamometer." This is a large, expensive piece of equipment found only in engineering laboratories and a few of the better equipped hot rod shops. But even though the average rodder can't measure his engine's torque output he should understand the formulas that apply to torque and horsepower.

Torque is measured in "pounds-feet." A pound-foot is a force of one pound exerted on a lever one-foot long. Thirty-three thousand pounds-feet of work done in one minute, no matter by what means, equals one horsepower. Because the force exerted by a flywheel or crankshaft acts in a circular direction, the constant "2 pi" is inserted in the formula. This makes the standard formula for the horsepower developed by an engine:

$$\frac{2 \text{ pi} \times \text{torque} \times \text{RPM}}{33{,}000} = \text{Horsepower}$$

By eliminating the 2 pi constant and reducing the 33,000 accordingly, the formula is reduced to:

$$\frac{\text{Torque} \times \text{RPM}}{5252} = \text{Horsepower}$$

As an example, let's compute the horsepower output of a 1960 Chrysler 300 F at its advertised maximum torque output of 495 pounds-feet at 2800 rpm. Applying these figures to the formula we get the equation:

$$\frac{495 \times 2800}{5252} = \frac{1{,}386{,}000}{5252} = 263 \text{ horsepower}$$

For computing torque when horsepower and rpm's are known, the formula is:

$$\frac{\text{Horsepower} \times 5252}{\text{RPM}} = \text{Torque}$$

To compute the torque delivered by the 300 F when it is developing its maximum advertised horsepower of 375 at 5000 rpm, the formula would be:

$$\frac{375 \times 5252}{5000} = \frac{1{,}969{,}500}{5000} = 393 \text{ pounds-feet of torque}$$

PISTON SPEED

In this age of short-stroke engines and sturdy pistons, piston speed in competition engines has ceased to be much of a problem. In the past, when engines had considerably longer strokes and weren't as well constructed as they are now, a piston speed of 3500 feet per minute was considered to be just about the safe maximum. Anything over this was apt to lead to disaster.

Computing piston speed is merely a matter of determining how far a piston moves in its cylinder during a minute of running time. The factors involved are the length of the stroke through which the piston moves and the crankshaft speed of the engine. Crankshaft speed is measured in revolutions per minute. The stroke length must be multiplied by 2 because the piston moves from the top to the bottom of its cylinder and back again to the top each time the crankshaft revolves one revolution. If the stroke length is quoted in inches, a factor of 12 is used in the formula to convert inches per minute to feet per minute.

The formula is:

$$\frac{\text{Stroke} \times 2 \times \text{RPM}}{12}$$

= Piston speed in feet per minute

For a 283 cubic inch Chevrolet engine with a standard crankshaft stroke of 3 inches running at 5000 rpm, the equation becomes:

$$\frac{3 \times 2 \times 5000}{12} = \frac{30{,}000}{12} = 2500 \text{ feet per minute}$$

STROKER PISTONS

Most fellows buy their stroker pistons now in kits that include a crankshaft of the correct stroke, connecting rods, and other necessary related parts for stroking an engine but before such kits became popular a rodder had to know how to order pistons to go with a stroked crankshaft.

A piston for a shaft that has a stroke longer than stock differs from a standard piston in that the distance from the center of its piston pin bore to the top of its head, which is its "compression height," is shorter to compensate for the greater distance it moves up the cylinder on compression and exhaust strokes. Most piston manufacturers designate their stroker pistons with a fraction of an inch and "L.C." stamped in the head of each piston. The fraction of an inch designates how much the compression height differs from that of a standard piston and the L.C. stands for "low crown" to indicate that the compression height is shorter than standard.

The formula for computing the L.C. dimension of pistons for any given stroke increase is:

$$\frac{\text{Stroke increase}}{2} = \text{L.C.}$$

For a ¼-inch stroke increase (.250-inch) the equation would be:

$$\frac{.250}{2} = .125\text{-inch } (⅛ \text{ inch}) \text{ L.C.}$$

The head of a ⅛ L.C. piston used with a ¼-inch stroked crankshaft would rise to the same position in a cylinder as the head of a standard piston used with a standard shaft.

Sometimes, for compression ratio purposes, it is desirable to have the head of a piston used with a stroked crankshaft reach a higher or lower position in the cylinder at the end of compression strokes. Computing the correct L.C. height for pistons for these conditions is simply a matter of adding the amount the piston head is to rise higher in the cylinder, or subtracting the amount the piston head is to be lower in the cylinder, to or from the standard L.C. height for the crankshaft being used.

VALVE LIFT

The only time a formula is needed for computing valve lift is when the lift is to be determined directly from the camshaft. Factors involved here are the heel diameter of the cams on the shaft and the height of the lobes on the cams. If the engine has overhead valves that are actuated by rocker arms, the rocker arm ratio must also be considered.

For an L-head engine, which is one in which the valves are actuated directly by lifters that contact the cams, valve lift is equal to the cam lobe measurement minus the heel diameter measurement minus the lash clearance measurement. The formula is:

Lobe—Heel—Clearance=Lift

For a theoretical camshaft that has a lobe measurement of 1.812-inch, a heel measurement of 1.500-inch, and a specified las' clearance of .012-inch, the equation would be:

$$1.812 - 1.500 - .012 = .300\text{-inch lift}$$

When rocker arms are involved, the ratio of the arms is added to the formula. Rocker arm ratios may be obtained

from factory engine specifications. They usually range between 1.5 and 1.8 to 1. This means that the valve opens 1.5 to 1.8 times the lift provided by the cams. The formula is:

Lobe measurement–Heel diameter×Rocker arm ratio–Lash Clearance=Lift

For an engine that has a cam lobe measurement of 1.750-inch, a cam heel diameter of 1.500-inch, a rocker arm ratio of 1.5 to 1, and a lash clearance of .015-inch, the equation would be:

$$1.750 - 1.500 \times 1.5 - .015 = .360\text{-inch lift}$$

VACUUM AND PRESSURE

Standard measures for vacuum and pressure are inches of mercury for vacuum and pounds per square inch (psi) for pressure; however, inches of mercury are often used when measuring positive intake manifold pressures created by blowers.

As one inch of mercury equals .49 psi, any measurement made in inches of mercury can be converted to pounds by multiplying it by .49. Conversely, any measurement made in psi can be converted to inches of mercury by dividing it by .49. Formulas for these are:

$$\text{Inches of mercury} \times .49 = \text{psi}$$

$$\frac{\text{psi}}{.49} = \text{Inches of mercury}$$

It must be remembered when dealing with pressures that usually their measurements are based on atmospheric pressure. Atmospheric pressure is the pressure of the air that covers the world's surface. Altitude, temperature, and other factors cause changes in atmospheric pressure but the figure used as standard for sea level and standard temperature is 14.7 psi. The equivalent in inches of mercury is 29.92 inches.

Pressure indicated by gauges of the type used in automobiles and for most other applications is the pressure above atmospheric. In other words, the Zero on the gauge is equivalent to atmospheric pressure. Pressure specified as "absolute" is the pressure above an absolute vacuum. On a gauge that indicates absolute pressure, atmospheric pressure is shown at its actual value. Pressure above atmospheric adds to the gauge's reading. Gauges of this type are often used to measure the manifold pressure in engines that have blowers.

A vacuum is a pressure below atmospheric. In a vessel in which there was a vacuum of five inches of mercury the actual pressure would be:

$5 \times .49 = 2.45$ pounds below standard atmospheric pressure.

FUEL MILEAGE

Hot rodders are seldom inerested in fuel mileage but those who are can easily determine the mileage their car is getting per gallon of fuel by using the formula:

$$\frac{\text{Distance driven}}{\text{Gallons of fuel used}} = \text{Miles per gallon}$$

For a trip of 210 miles that required 10 gallons of fuel the equation would be:

$$\frac{210}{10} = 21 \text{ miles per gallon}$$

For the results of this formula to be correct it is necessary that both the mileage and the amount of fuel used be measured accurately. A slight inaccuracy in either measurement can cause the result to vary considerably.

UNITED STATES-METRIC MEASUREMENTS

With so many cars and motorcycles of foreign manufacture being imported into this country these days it is important for a rodder who works on this equipment, or merely reads about it, to have an understanding of the relationship of Metric measurements to the measurements used in this country. He must know how to convert one to the other if he is to be able to compare the foreign machines with ours.

Bore and stroke measurements in foreign engines are usually stated in millimeters and displacements in cubic centimeters. One cubic centimeter equals 1000 cubic millimeters.

The formula for converting bore and stroke measurements from millimeters to inches is

$$\frac{\text{Millimeters}}{25.4} = \text{Inches}$$

A cylinder bore of 76 millimeters has a diameter of

$$\frac{76}{25.4} = 2.99 \text{ inches}$$

Inches are converted to millimeters with the formula

$$25.4 \times \text{Inches} = \text{Millimeters.}$$

In millimeters, a 3¼-inch piston stroke would be

$$25.4 \times 3.250 = 82.45 \text{ millimeters}$$

As one cubic inch equals 16.387 cubic centimeters (cc), the formula for converting a displacement in cc's to cubic inches is

$$\frac{\text{cc's}}{16.387} = \text{cubic inches}$$

For example, consider an engine of 1000 cc's:

$$\frac{1000}{16.387} = 61.023 \text{ cubic inches}$$

For cubic inches to cc's, the formula is

$$\text{Cubic inches} \times 16.387 = \text{cc's}$$

A 283 cubic inch Chevrolet V8 engine has

$$283 \times 16.387 = 4764.87 \text{ cc's}$$

To keep the numbers from becoming so large, displacements are often stated in "Liters" rather than cc's, such as 2 liters, 3.8 liter, etc. One liter equals 1000 cubic centimeters, or 61.023 cubic inches. The conversion formula for liters to cubic inches is

$$\text{Liters} \times 61.023 = \text{cubic inches.}$$

For converting cubic inches to liters, use the formula

$$\frac{\text{Cubic inches}}{61.023} = \text{Liters}$$

For example, the displacement of a 409 Chevy in liters is

$$\frac{409}{61.023} = 6.7 \text{ liters}$$

Sometimes car weights are specified in "cwt," which is the abbreviation for "hundredweight." In England, where the term is used, 1 cwt usually equals 112 avoirdupois pounds; therefore, in this country it also equals 112 pounds.

To convert cwt to pounds, use the formula

$$\text{cwt} \times 112 = \text{Pounds}$$

A car that was said to weigh 14 cwt would weigh

$$14 \times 112 = 1568 \text{ pounds}$$

The formula for converting pounds to cwt is

$$\frac{\text{Pounds}}{112} = \text{cwt}$$

A Ford that weighs 3600 pounds weighs

$$\frac{3600}{112} = 32.14 \text{ cwt}$$

Speed and mileages stated in kilometers can be confusing unless a rodder knows that

1 kilometer = .6214 mile, or that
1 mile = 1.609 kilometers

To convert kilometers to miles, use the formula

$$\text{Kilometers} \times .6214 = \text{Miles}$$

A distance of 90 kilometers, or a speed of 90 kph, becomes

$$90 \times .6214 = 54.29 \text{ miles, or mph}$$

To convert miles to kilometers, use the formula

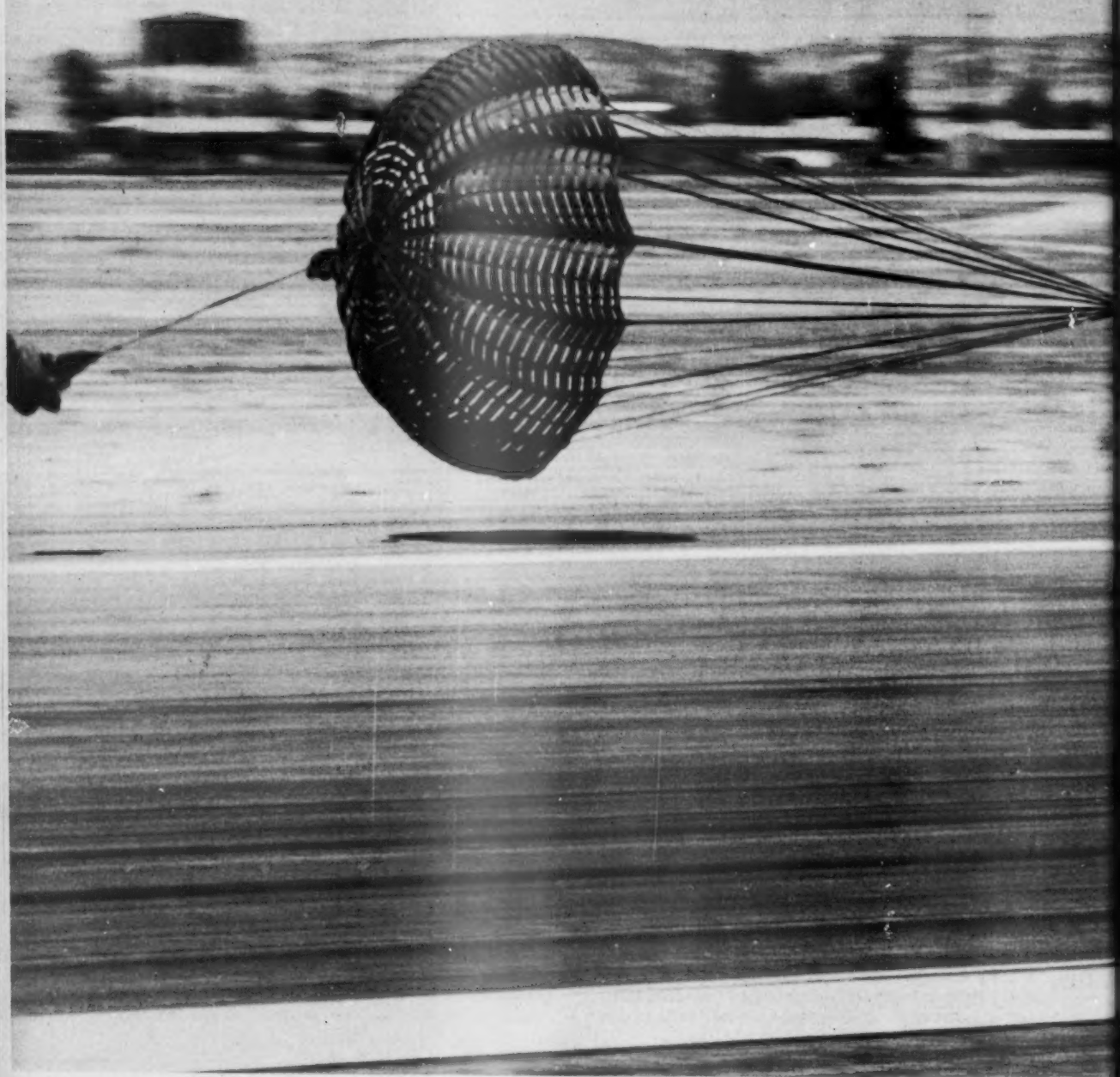
$$\text{Miles} \times 1.609 = \text{Kilometers}$$

A distance of 80 miles, or a speed of 80 mph, would be

$$80 \times 1.609 = 128.72 \text{ kilometers, or kph.}$$

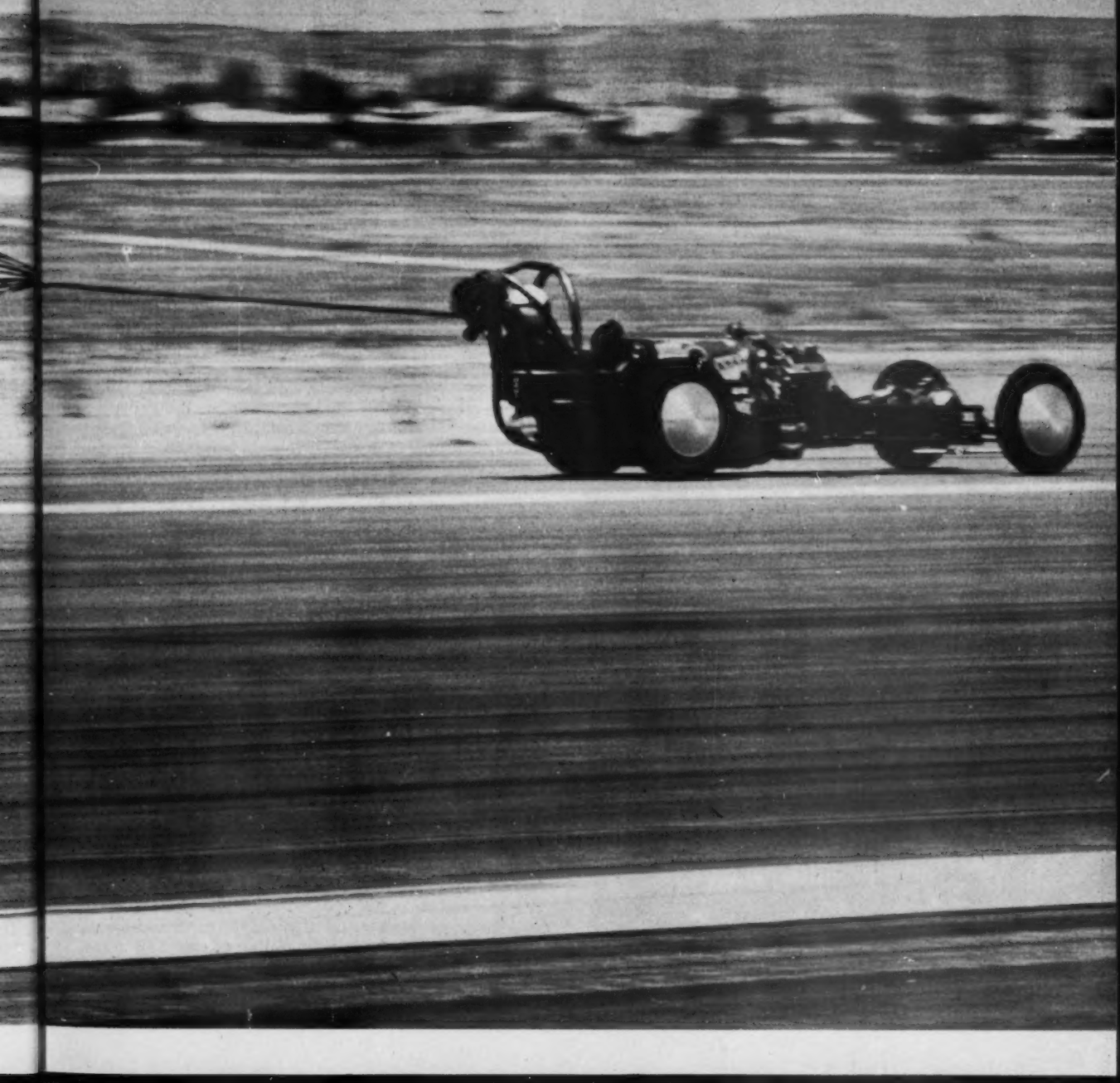
DON FRANCISCO

DRAG CHUTES FOR DRAG CARS



WHEN drag racing first became an organized sport the strips on which events were held were adequate for the cars that were running. They had a measured quarter-mile for accelerating and as much and possibly a little more for stopping. At the speeds cars were running none of them had trouble stopping in the distance provided. But then came the dragsters. Speeds over the same quarter-mile acceleration distance jumped fantastically but the length of the stopping area remained the same. Stopping a car became as much an art as getting it going. Few fellows actually had trouble stopping, although brakes and tires didn't last long, but there was always the chance that something might happen to the brakes or some other critical part of a car that would put its driver in the precarious position of not being able to stop. This could lead to disaster.

Now the dragster driver's lot is an easier one. The drag chute his car wears on its back, as a hiker wears a knapsack, is his insurance for a fast, safe stop regardless of how fast his car may go. An additional advantage of the chute is that it will usually straighten the car if it should get into trouble by trying to veer off the strip, turn broadside, or assume any of the attitudes a high-speed car can get into when something unexpected happens. Several drivers owe their lives to the chute that held their car straight and



Jim Deist displays one of the aircraft "ribbon" drag chutes he reworks for dragsters and other types of drag racing machines. Ribbon chutes were a German development.

This is a "ring slot" drag chute, an American improvement of the ribbon chute. Size for size, ring slot chutes have a better drag coefficient than ribbon chutes.

Displayed at the right on Jim's packing table is the standard towline for a ring slot. At left is Jim's lightweight line.

stopped it when some emergency arose. But if this gives you the idea drag chutes are for emergencies only, forget it. Drivers of most of the country's faster dragsters use their car's chute every run, just as they use its throttle, clutch, and brakes.

It's easy to understand why anything as large as a drag chute trailing behind a dragster will cause the car to decelerate rapidly but the thing about a chute that causes it to snap a car that has started to go somewhere other than straight down the strip back into alignment with the strip is a little more difficult to comprehend. The straightening effect is the result of the chute's deceleration force being in line with the strip. In other words, the car's direction of movement is returned to the direction in which the chute's towline is pointing. This characteristic doesn't cause any safety problem that might arise from a short course or one that has a curve at its end because once the chute is open and the car is traveling straight the chute will follow when the car is steered.

Drag chutes are an offspring of the jet age. They differ from the parachutes that more or less gently ease a man who steps from an airplane back to earth by having only partial instead of full canopies. Also, chutes used for drag cars are smaller in diameter than those used by flyers and sport jumpers.

Drag chutes were designed originally as deceleration aids for jet fighter planes that had to land on short fields. Now, modern military jets of all types use them as a matter of course when landing. Sometimes its chute is opened while a plane is in the air and other times not until its wheels are solidly on the ground. The drag created by the chute slows the plane and helps its pilot maintain directional control. When the plane slows to taxiing speed the chute is dropped so it won't be damaged by being dragged on the runway or taxiway.

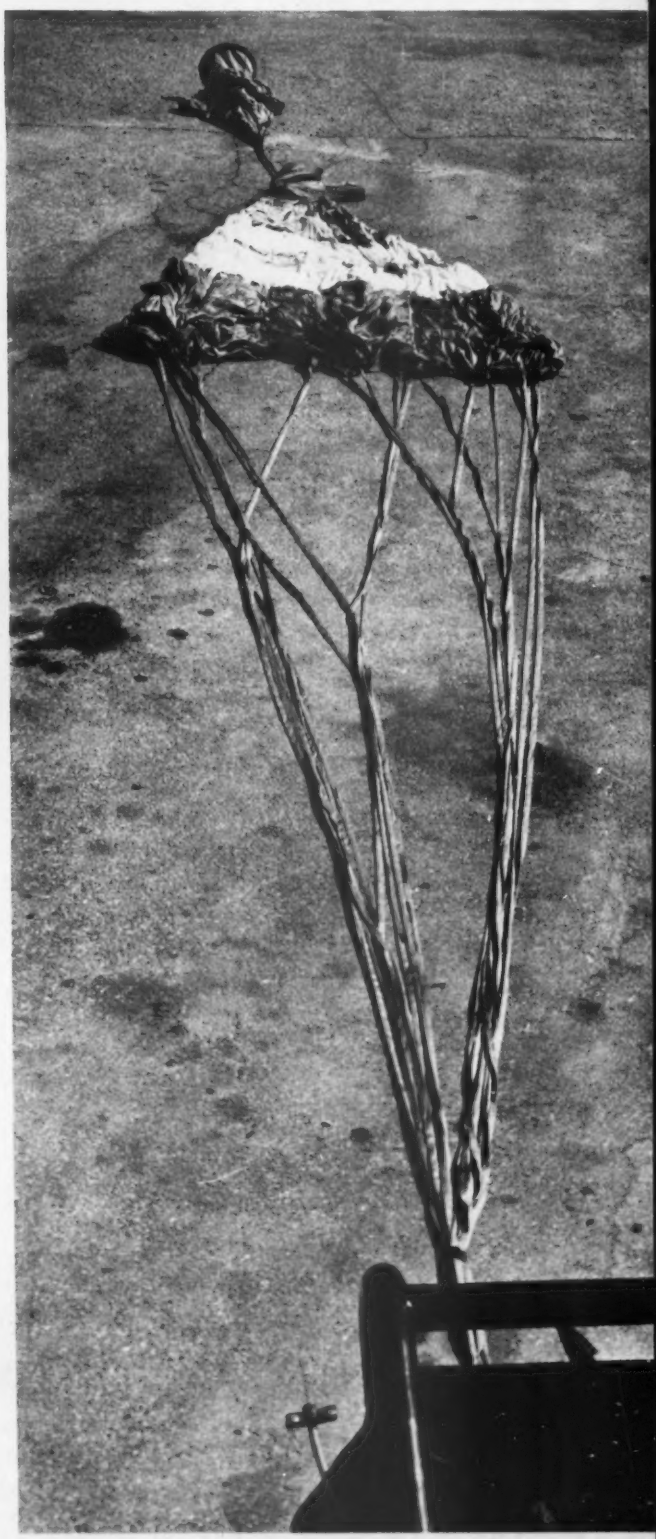
The use of drag chutes isn't limited to dragsters. This one helps stop a hot sedan. Note the towline's attachment point.

Two types of drag chutes are the most popular for dragster use. One of these, which is called a "ribbon" chute, was the first to be used on aircraft. It is of German origin. It consists of a network of two-inch wide nylon ribbons that have a tensile strength of approximately 300 pounds. The second is an American improvement of the ribbon chute. It is the "ring slot," which consists of concentric rings of nylon cloth eight to fifteen inches wide sewed to radial "tapes," which are nylon straps. The tapes hold the rings in place and receive the pressure exerted on them by the air through which the chute is moving.

The various parts of a chute assembly are its pack, release cable, pilot chute, main chute, and towline. Packs are made in different types but they all serve the purpose of being a container for the main chute, the pilot chute, and the towline. Four flaps on the rear of the pack that act as a cover to hold the chutes in place are held closed by one end of the release cable. The cable passes through a nylon loop attached to the top of the pilot chute and which is threaded through eyelets in the flaps.

Enclosed in the pilot chute is a long coil spring that is compressed when the chute is packed. This spring ejects the pilot chute from the pack when the release cable is

It is the duty of this pilot chute and its coil spring to eject the main chute from its pack when the car's driver pulls the chute's release ring.



After a dragster has come to a stop, its chute looks something like this. Now the driver and his crew have to repack the chute for the next run. This isn't too difficult.

Packing a chute is a two-man job. Here, Jerry Moffatt holds the chute while Jim separates its shroud lines. This is step one.

Step two is for the shroud man to get between the shroud lines, move down to the chute to separate the canopy into halves.

With one set of shroud lines on his shoulder, Jim begins gathering and folding the panels of the canopy for the other set.

pulled. Elastic straps connected to the pack's flaps pull the flaps to their open position to make it easier for the pilot chute to eject. A nylon strap, called a "bridle," connects the pilot chute to the main chute.

Parts of the main chute are the canopy, which is the member that catches the air; the nylon suspension lines, sometimes called "shroud" lines; and the nylon towline. The suspension lines connect the towline to the canopy; their length depends primarily on the chute's diameter. Towline length is engineered for the car.

Measures of a drag chute's efficiency are its "opening shock" and its "steady state drag." These factors are forces, rated in pounds, the chute exerts on its towline. Opening shock is usually about 1.05 times steady state drag. These factors, for chutes of different types and diameters, are listed in the accompanying chart.

Drag chutes for cars were practically an unknown quantity prior to 1958. A chute of some sort had been seen on a sporty car that ran at Bonneville in '57 but no one seemed to know how well it had worked. When a dragster owner asked Jim Deist, who was then working for Irving Airchute Co., if a chute would help stop his dragster, Jim didn't know. He had designed and manufactured chutes for men and drag chutes for planes and missiles but he didn't have the answer to this question. To find the answer, he did some experimenting with a chute connected to a passenger car. It was these experiments that aroused his interest in the possibilities drag chutes held for drag racing cars to the extent that he quit his job at Irving and started his own business of designing and manufacturing them. This was the first company to specialize in chutes of this type. Jim calls it Deist Drag Chute Co. It is in Glendale, Calif.

When Jim first began making drag chutes he knew that a ring slot had a better drag coefficient than a ribbon chute of the same size but it wasn't until he had been in the business about a year and a half that he decided to run

Panels gathered and shroud lines in hand, the lines are placed on the shoulder and the operation repeated for the other side.

After both sides of the chute have been gathered, both sets of lines are transferred to one hand. Note that the lines are kept separated with the fingers.

LEFT—The shroud lines are shifted to Jerry, who holds them with his right hand in such a manner to keep them separated.

Jim threads the lanyard that will be used to pull the loop on the pilot chute's end through the pack's flap through the loop.

some tests with a ring slot to determine whether it would withstand the rigors of drag racing as well as the ribbon chutes were. After replacing the ring slot's standard towline with a line of lighter weight he found it had very good wear characteristics. So good, in fact, that now his best chute is a ring slot.

The principal idea of replacing the towline on a ring slot chute is to reduce the chute's weight. Reducing its weight lowers the speed required to keep the chute in the air. As the thing that causes a chute to wear is its being dragged after it has collapsed, it is apparent that the longer a chute will remain in the air when a car is slowing to a stop the less wear it will suffer. The actual weight reduction Jim achieves in a fourteen and a half foot diameter ring slot by changing its towline is six and a half pounds. This is quite a bit in comparison to the chute's original weight of sixteen pounds, ten ounces.

A driver should use his car's chute on every high-speed run. If he does this, pulling the ring on its release cable will become an automatic action. Then, if anything out of the ordinary that might get his car into trouble should happen on a run, the driver will release the chute automatically. This will give the chute a chance to open before the car spins or gets out of control some other way. The chute will then help keep the car straight on the course and enable its driver to stop it in the shortest possible distance.

For normal stops, when an emergency doesn't exist, Jim recommends the chute release be pulled as the car approaches the first timing light. This would seem to be much too soon but from close observation of drivers actually pulling the release it was learned that a driver's reaction time is slow enough that when he tries to pull it at the first light the chute doesn't come out of its pack until the car is approximately a hundred feet past the last light. This is ideal as it isn't early enough to cut the car's time nor so late that too much of the shutoff distance is used getting the chute open. The distance from the point where the ring is pulled to where the car stops is usually about seven hundred feet.

To shorten the distance a chute is dragged Jim recommends that it be allowed to slow the car to a speed somewhere between fifty and twenty-five miles per hour. It won't touch the strip as long as it is inflated because the air spilling out of its lower side acts as a cushion to hold it up but somewhere between fifty and twenty-five miles per hour it will start to deflate. Then, the car should be turned in the direction of the return road and quickly braked to a stop. When a car is stopped in this manner its chute will drop to the strip and remain where it touches. If the car isn't stopped quickly enough the chute will be dragged. Dragster owners estimate that this method of stopping eliminates as much as sixty percent of the wear a car's rear tires and brakes get when brakes alone are used.

Jim can supply a chute for any type of drag car but his most popular model is the fourteen and a half foot diameter ring slot. This is usually recommended for cars under fifteen hundred pounds, depending on the car's maximum speed. For speeds under approximately 175 mph, possibly a sixteen-foot chute should be used. A sixteen-foot ribbon type would do the job but a ring slot of the same size, with Jim's new towline, would stop the car a little more easily. The different steady state drag ratings of the two types, as shown on the accompanying chart, prove the ring slot to be the better of the two.

The reason car speed has an influence on the size chute a car should have is that the drag a chute exerts becomes greater as speed increases. The rate at which drag goes up is quite pronounced. A general rule is that the faster and lighter the car, the smaller the chute can be. Jim can calculate the size chute that would be the most practical for any car.

This is the empty pack, ready for the chute. Note how the towline passes through an opening in the forward side of the pack.

After replacing broken or missing rubber bands, the towline is folded, secured with the bands, working up from the bottom.

As the towline is being folded into the pack, the man holding the chute moves forward until he is directly behind the car.

The canopy is then folded into the pack in the same manner as the towline and shroud lines, but there is nothing to hold it.

As the folding of the canopy continues, the man who is holding the canopy helps the other man by holding one end of the folds.

Folding and stuffing of the canopy is continued until all of it is in the pack, like ten pounds of stuff in a two-pound bag.

The pilot chute separator flap covers the main chute to prevent the pilot chute and its bridle from becoming tangled with it.

The lower end of the pilot chute is gathered together around the chute's spring while the other man holds the main chute.

The smaller chutes Jim sells are for Bonneville cars or ultra-light dragsters. According to the rule they would be for cars of average weight capable of very high speeds, or for very light but slower cars.

A car fitted with a drag chute must have a good seat belt and shoulder harness arrangement that will hold its driver securely in his seat when the chute opens. Jim estimates that a sixteen-foot ring slot on a fifteen hundred pound car could create sufficient drag to subject the car and its driver to six g's of deceleration force. When a man jumps out of an airplane he receives a shock of eight to ten g's when his chute opens. Jim has done considerable research to determine the number of g's resulting from deceleration a car's driver can withstand while in a sitting position without too much discomfort or blacking-out but he has been unable to find any specific information on the subject; however, he knows the number is considerably more than six.

Jim controls the number of g's a chute exerts on a car when it opens by varying the size of its pilot chute. A fairly large pilot chute will delay the opening of the main chute by pulling so hard on its center that it can't inflate to its full size as quickly as it would otherwise. However, while the main chute is only partially inflated, or in a "reefed" condition, it is still exerting a drag on the car. This reduces the car's speed so that when the main chute does inflate to its full diameter the deceleration force it exerts on the car and driver is considerably less than it would have been if the chute's inflation hadn't been delayed.

Jim has found that most drivers want to feel the chute when it opens. They don't want it to open so abruptly they'll get black and blue marks from their shoulder harness but they want to feel the deceleration. He estimates, from descriptions of the force drivers have given him, that most of his installations are exerting a maximum opening shock of approximately three g's. Three g's will stop a car quickly but not so abruptly that its driver will suffer any discomfort.

Installation of a chute is a simple matter of mounting its pack on the rear of the car so it won't fall off during acceleration and providing a sturdy attachment point for its towline. All the pack does is carry the packed chute. The drag of the open chute is exerted on the towline attachment point, which should be capable of holding a ten thousand to twelve thousand pound pull. There must be a certain safety factor here so that if the car should make a lucky run some day and go considerably faster than its usual speed the attachment point won't be jerked loose from the car.

The attachment point must be somewhere on a vertical line that is midway between the car's rear wheels. Jim recommends it be approximately ten inches above the rear axle's centerline. A high attachment point eliminates the tendency of the chute to lift the car's rear wheels off the strip. Some drivers swear that the rear end of their car is lifted by the chute but this is merely an effect of the g force their body experiences when the chute opens. As they rock forward in their seat against the seat belt and shoulder harness they get the feeling that the rear of the car is being lifted. This feeling can be accentuated if the driver is exerting pressure with a foot on a clutch pedal. The pedal acts as a fulcrum point about which his body will tend to rotate.

The chute should be mounted fairly high on the car. This makes packing it easier but more important than this is that the air at a higher point is not as likely to be as turbulent as at a lower point. Sometimes it helps to mount the chute on a slight angle, with its upper end forward.

Turbulence in the air behind a dragster can be caused by wide slicks, the bulk of the engine, and even the driver's helmet. The problem it creates is lack of enough air to grab the pilot chute when it is ejected from the pack. As a result, the pilot chute will hesitate, causing a loss of time before it pulls the main chute out of the pack. This loss of time can't be tolerated with the short stopping distance most drag strips have.

The chute's release cable can be installed so that its D-ring is in any location the driver prefers. This is usually convenient to his free hand. If he steers with his right hand and operates other controls with his left hand, the D-ring should be on the left side of the cockpit, and vice versa. For drivers who like to keep both hands on the wheel at all times, the release can be set up so that it can be operated with a foot. Jim recommends that movement of the D-ring required to release the chute be from two to four inches. This is determined by the length of release wire extending through the loop that secures the cover's flaps. The wire and its housing are easily shortened to the desired length.

To the average hot rodder the most mysterious thing about a chute is repacking it after it has been used. Just

While holding the pilot chute's lower end, the chute's spring is compressed by forcing the chute's outer end toward the flap.

This is where the lanyard comes into action. Both of its ends are threaded through the grommet in the upper flap of the pack.

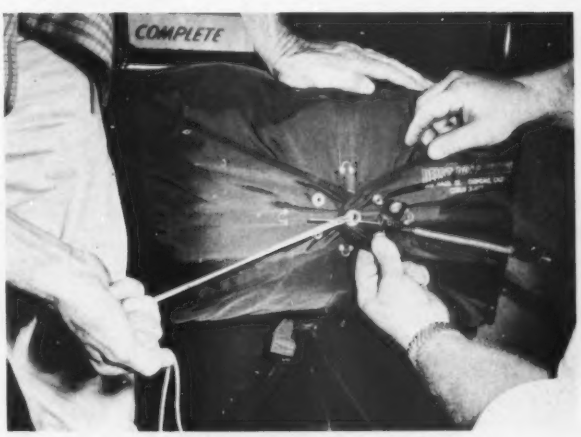
The loop on the pilot chute is pulled through the grommet in the upper flap, then through the lower flap in the same way.

After the upper and lower flaps have been secured, the lanyard and loop are threaded through the grommets in the side flaps.

how do you get all that stuff back into that little container so it will come out again as it should? According to Jim, this is simple. Two fellows who have done the job a few times and worked out a system can pack a chute in about three mintues. Two fellows who are all thumbs and very green at the task shouldn't take over ten minutes. At the present time Jim takes the new owner through the job step by step, if this is possible. If the chute is shipped to a customer, Jim recommends that the owner carefully unpack it when he receives it, observing as he goes how the parts are stowed in the pack.

Jim has found that at the points where a fellow would hesitate while packing a chute because he didn't know what to do, usually it doesn't matter what he does. As long as the pilot chute is on the outside of the main chute so its spring can eject it from the pack when the release cable is pulled and the lines are stowed in an orderly fashion, the packing job is usually o.k.

The best way to take care of a chute is to use it. Driving a car run after run without popping its chute may cause the continual jolting the chute receives to pound it down



While the lanyard is still in the loop, the release wire is inserted in the loop and its housing is snapped to the cover.

Drag Forces Created by Deist Chutes		
All Forces in Pounds		
Chute opened at 175 mph (257.25 feet per second)		
Type and size	Opening shock	Steady State Drag
16-foot ring slot	9130	8690
16-foot ribbon	8300	7904
14½-foot ring slot	7495	7135
12-foot ring slot	5120	4880
12-foot ribbon	4655	4445
11-foot ribbon	4095	3900
Chute opened at 200 mph (293 feet per second)		
16-foot ring slot	12,330	11,770
14½-foot ring slot	10,150	9665

into its pack and become tangled. When a chute is in this condition it's possible that it won't work as it should when it is needed.

The useful life of a chute that isn't dragged excessively is phenomenal. After the Deist chute on one popular drag racer's car had been used approximately four hundred and fifty times it was returned to Deist for a checkup. The bill for putting it back in good condition was thirty dollars. This is a little over six cents per run, which is an insignificant premium for the insurance a chute provides.

If a chute is to be stored for a long period of time, such as for the winter, don't leave it packed. It seems that the nylon in chutes loses some of its flexibility if it is stored in a tightly compressed condition for any length of time. Take the chute out of the pack, roll it and the towline into a loose bundle, and place it in a paper bag or similar container. Be sure the pilot chute spring is released so it won't take a permanent set. Also be sure the chute is dry. Water or moisture won't hurt the nylon canopy and lines but they could cause foreign matter that collected on them during normal use to mildew. This isn't good. Store the bundled chute in a dry place, preferably one that is warm.

Prices of Deist's chutes range from seventy-five to two hundred and fifty dollars, depending on what the customer wants. The seventy-five dollar unit is a new surplus ring slot and pack assembly. It is complete, with pilot chute and towline. A one hundred dollar chute is a Deist "Custom Pack" with a surplus ring slot canopy.

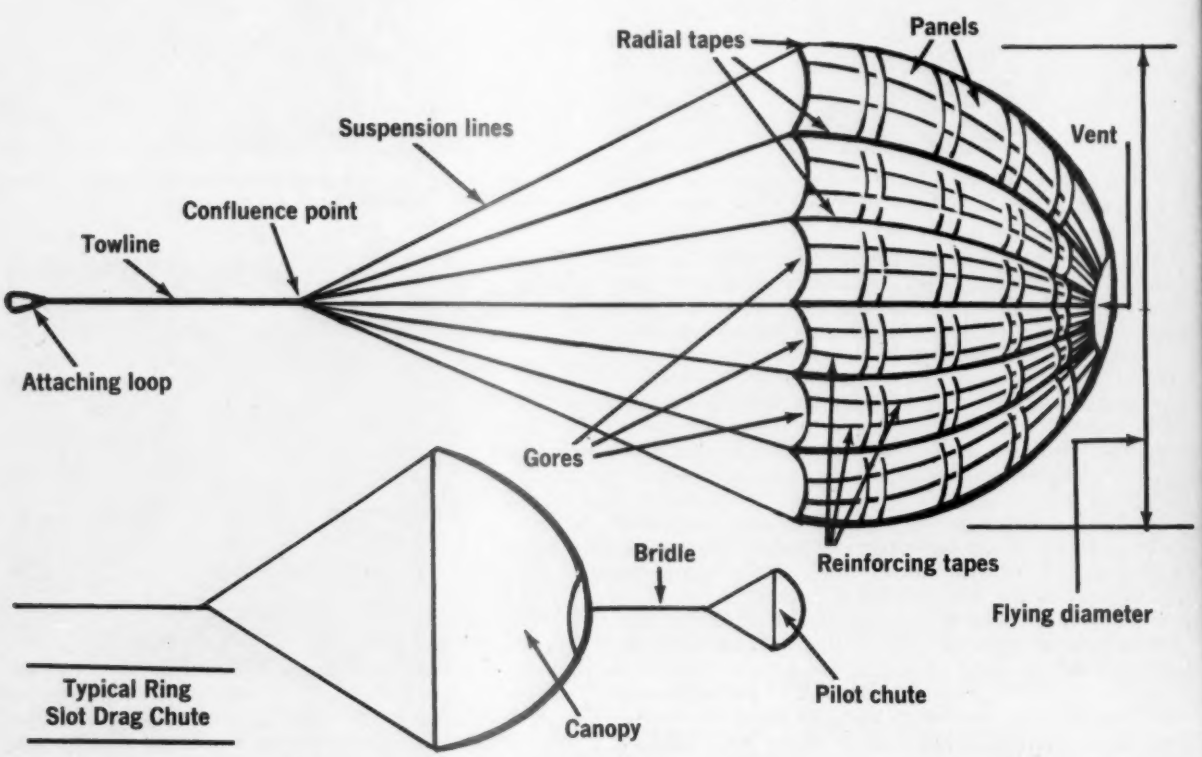
Jim considers his one hundred and thirty-five dollar chute to be the best buy for the average car. It is a sixteen-foot used surplus ring slot canopy that has been reconditioned and fitted with a light-weight towline. A new pilot chute and a Custom Pack are included. From this point on the price depends on what the customer wants. He may want a new canopy, a larger canopy, a special towline, special dye job, etc.

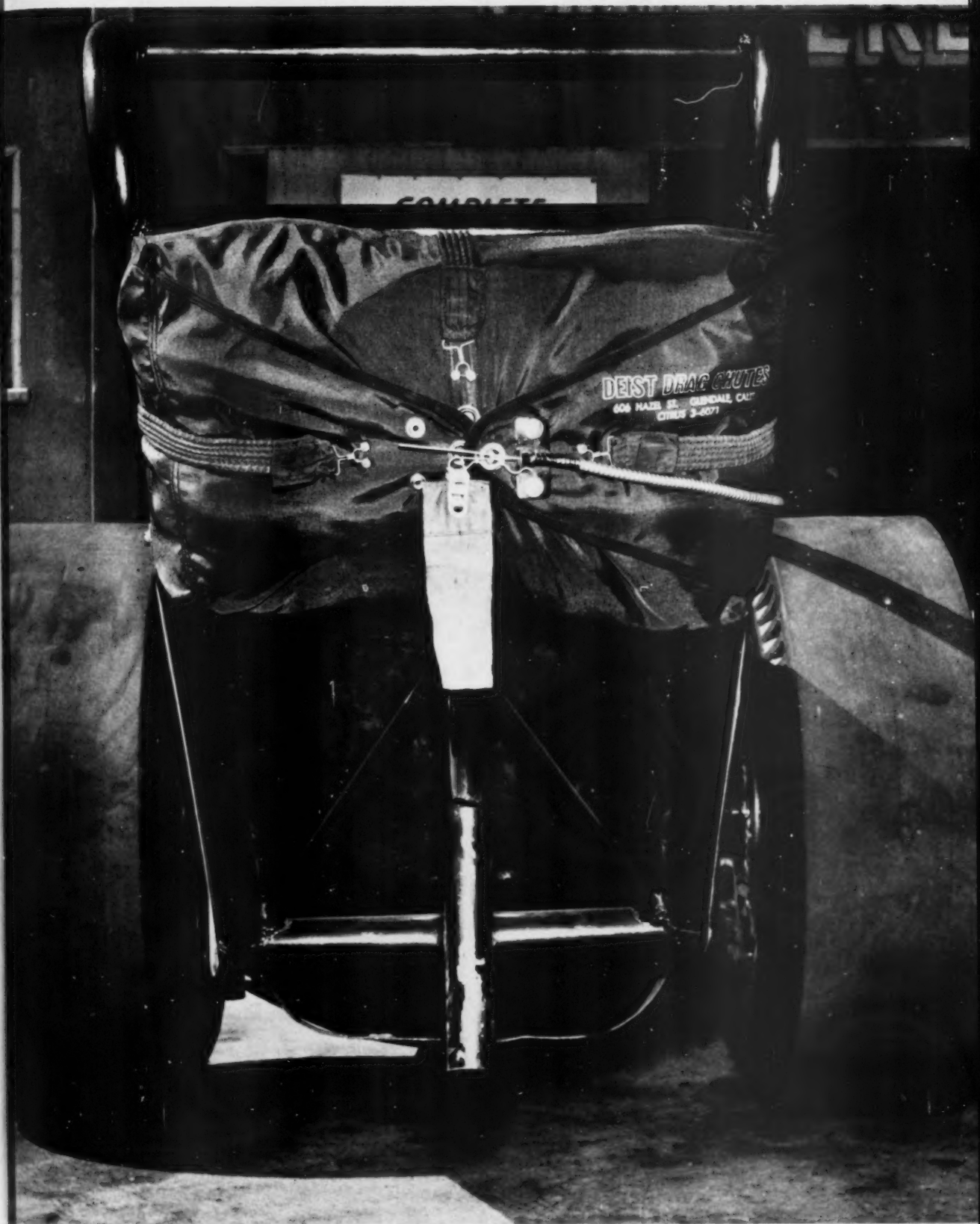
Installation of dual or twin chutes is not recommended because of many difficulties involved in getting multiple chutes to function correctly. However, if a single chute can't do the job it may be possible that Jim can design a dual setup that will.

For weight conscious car owners, a single chute setup weighs fourteen to eighteen pounds, complete, depending on its size.

If you're planning on installing a chute on a drag car, remember this: Just any chute won't do. The chute has to be engineered to the car. A few fellows have learned this the expensive way. They have bought chutes from surplus stores and other sources and gone to the trouble of installing them only to find they were the wrong size or the wrong type or that something else was wrong with them. It's fellows like this who can give chutes for drag cars a bad name. Chutes engineered for the job by someone like Jim Deist will do the job.

DON FRANCISCO





The job is finished by attaching the release springs to the flaps and inserting a safety pin attached to a red ribbon in the loop.

ALUMINUM CONNECTING RODS

AMONG the latest of the new engine parts to start flowing from the shops of men who manufacture hot rod equipment are aluminum connecting rods. The purpose of aluminum rods is to help an all-out competition engine stay together rather than improve its horsepower output.

The steel connecting rods that are standard equipment in late model engines are strong enough for stock engines and even those that have been reworked quite extensively but short life of the crankpin bearings used in them and frequent breaking proved some time ago that they weren't adequate in engines fitted with big 6-71 GMC blowers or that were operated on highly nitrated fuels.

A summation of the theories held by manufacturers of aluminum rods for the inadequacies of standard rods in high-output engines are (1) That normal combustion pressures in reworked engines that deliver the very high horsepower outputs common today are too high for standard rods to support. (2) That combustion pressures resulting from detonation, which can be much higher than normal combustion pressures, overload the rods and cause them to bend. (3) That high loads and high engine speeds cause the material in the rods to fatigue quickly and break in the areas where pressure moves from one side of a rod to the other each time a piston reaches the end of a stroke and reverses its direction. The area of the rods where breakage of this sort usually occurs is just below the piston pin boss. (4) That short crankpin bearing life and bearing insert spinning in rod big-end bores are results of the bores' becoming oversize and out of round when heavy loads are exerted on them.

Regardless of whether the manufacturers' theories are correct or not, the facts remain the same. These are that standard rods used in highly-stressed competition engines do bend and break, that the bearing inserts in their big-end bores sometimes spin in the bores, and the useful life of inserts that don't spin is quite short. A great amount of thought wasn't required to determine that connecting rods were another engine component that could stand improving.

The record shows Ed Iskenderian, of Iskenderian Racing Cams, to be the first of the competition engine parts manufacturers to try the aluminum rod solution to the connecting rod problem. This was in 1958. As Ed was interested in several Chrysler-powered drag cars at that time, his first rods were for Chrysler engines. These differed from the rods now being manufactured by Ed and other companies by being machined from aluminum flat stock rather than from forgings. They did the job they were supposed to do but they cost so much to produce that many rodders who needed them couldn't afford them. To cut production costs Isky switched to the forging process, and to cater to the greatest demand he discontinued the Chrysler rods and began making rods for the small Chevy V8's.

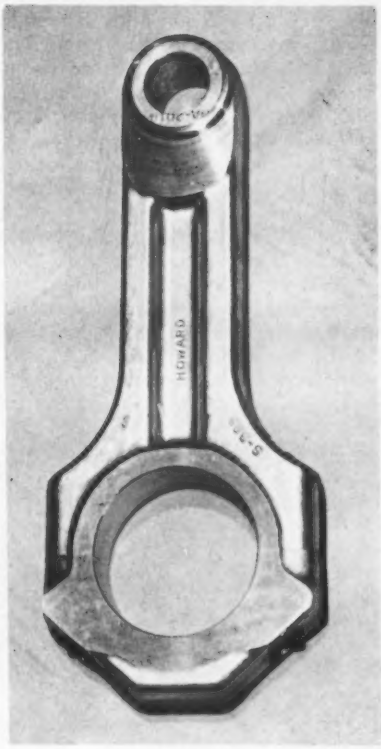
Other manufacturers soon joined Isky in the forged aluminum rod business. At the present time at least five of them are involved. Design details of the rods produced by these manufacturers vary, sometimes considerably, but the aims of all the manufacturers are to make the rods' big-ends strong enough to stay round and within an acceptable tolerance of the correct diameter at all times and their shanks strong enough to withstand any load an engine could possibly exert on them.

The first thing noticed about an aluminum rod is that it is so much bulkier than a steel rod for the same engine. Its shank is much wider and thicker, the boss for its pin bore is much thicker, and there is much more material around its big-end bore. This bulkiness, which doesn't interfere with the rods' function or their installation because there is plenty of clearance in the engine for it, is the reason for the rods' success. Because of the comparatively light weight of the aluminum alloys used, compared to that of steel, more of it can be used in a rod without exceeding the standard rod's weight. Weight is an important factor.

Although when compared on a cubic inch per cubic inch basis the aluminum alloys used for forged rods aren't as strong as steel, their strength, on an ounce per ounce basis, is much greater because an ounce of any of them has a much greater volume than an ounce of steel. Therefore, a rod made of twenty ounces of one of these alloys is much stronger than an equivalent rod made of twenty ounces of steel, if the material is correctly distributed over the rod's structure. One manufacturer estimates, on the basis of tests he has made, that his forged aluminum rods are four times stronger than the standard rods they replace, although they are a few grams lighter.

Another feature of aluminum rods their manufacturers say helps bearing life is due to aluminum's excellent heat dissipation qualities. Heat transferred to them from their crankpin bearings travels away from the bearings much more quickly than it does in steel rods. The heat is carried away from the rods by the continual bath of oil in which they move. The cooler a bearing can run, the longer it will run without failing.

Distribution of the material over a rod's structure is the problem of the man who designs the rod. Once he has



ABOVE, LEFT—Forged aluminum rods by Auto Dynamic Balancing have thread inserts, for most U.S. OHV, foreign engines.

ABOVE, MIDDLE—Arrow indicates 1/8-in. offset shank on ADB's rod that gives clearance for balancing plates added to cranks.

ABOVE, RIGHT—Two types of forged rods are made by Howard's. Length can be adjusted for big V8 engines.

RIGHT—Isky forged rod for Chevrolet V8 has offset cap that is said to allow more clearance for rods used on stroked shafts.

RIGHT, BOTTOM—Howard rod for small Chevy's. For use with long strokes, corners of the cap are clipped for block clearance.

selected the alloy he wants to use he has nothing to do with this part of the manufacturing process because the alloys are the products of some of the country's largest aluminum producers. Also, the aluminum companies do the forging and heat treating. Some of the manufacturers do the finish machine work on their rods but others have this part of the job done by machine shops better equipped to handle it.

One of the most prominent differences one will find in the aluminum rods now available is in the design of their big-end cap. On most makes the cap is of conventional design, which means that the parting line between it and the rod forms a ninety degree angle with the vertical centerline through the rod's shank. The parting line for caps on rods of other makes forms an angle of approximately forty-five degrees with the parting line on conventional rods.

In the days of small cylinder bores and long piston strokes rods with offset caps were often necessary to enable the rod big-ends to pass through the cylinders for rod and piston installation. Offsetting the cap makes the lower end of the rod, after the cap has been removed, narrower

than it would be otherwise. But with the monstrous cylinder bores that are common in modern engines, rod big-ends could be much larger than they are on any of the forged rods and still pass through the cylinders without difficulty. The reason one manufacturer gives for the offset caps on his rods is that the design provides more clearance between the lower ends of the rods and the cylinder block when the rods are installed on stroked crankshafts. This may be so; however, aluminum rods with caps of conventional design are installed on stroked cranks that have all practical stroke lengths without making necessary unusual modifications to the cylinder blocks in which they are used.

Some manufacturers use studs, which are screwed into the rods and fitted with nuts, to retain the caps and others use capscrews of one type or another. Each manufacturer has his reasons for his decision and both methods are being used successfully. One important rule that applies to both studs and capscrews is that they be strong enough to retain the cap without breaking and that they keep the cap correctly aligned on the rod.

Important details of either steel or forged aluminum connecting rods are their piston pin and big-end bore diameters, their center to center length, and their alignment. For both steel and aluminum rods these are determined when the pin and big-end bores are machined but the two types of rods differ in that errors in the alignment between the bores in steel rods can be corrected by twisting or bending the rods. Aluminum rods cannot be aligned in this manner because their strength makes it impractical to try to twist or bend them.

The diameter of the piston pin bore determines the clearance between the bore and the pin used in it. As all aluminum rods are designed for floating piston pins, their pin bores must be approximately .0009-inch larger than the pins. The rod material is used as the bearing surface. In steel rods designed for floating pins the pins ride in bronze bushings.

When floating pins are installed in rods and pistons designed for pressed-in pins it becomes necessary to have grooves machined in the ends of the pistons' pin bores for lock rings that will retain the pins in the rod and piston assemblies. This doesn't present any problem when aluminum rods are used because pistons for engines that require these rods are readily available with lock ring grooves.

The diameter of a connecting rod's big-end bore determines how the bore will grip the bearing inserts installed in it and the clearance the inserts will have on the crankshaft's crankpin. If the bore is oversize, the diameter of the bearing surface will also be oversize.

Piston pin and big-end bores in rods of any type must be round. The specified clearance between a piston pin and its bore can't be established if the bore is out of round because it will vary with the out of round condition. Although the bearing inserts installed in the rods have a steel backing, they are much more flexible than the big-ends. This means that when they are clamped in a rod they will conform to the shape of the big-end's bore. Inserts in an out of round bore will also be out of round. Correct bearing clearance can't be established if the bearings aren't round.

Careful machining is required if the pin and big-end bores in aluminum rods are to be round. The problem is the tendency of aluminum to expand when heated. Localized heat created by cutting tools when deep cuts are attempted distorts the area being machined. When this happens during the machining of pin and big-end bores, the result is out of round bores. The solution is to do the machining in several steps, taking a light cut each step. The bores should be finished to size by removing a thin layer of material from their surfaces with a hone. Honing is essential to accurate bore sizing and roundness.

It is also important that the other machining operations necessary on the rods, which are separating the cap from the rod, boring the rod and cap for the cap retaining bolts or studs, counterboring the caps for the bolt heads or stud nuts, milling the insert lock grooves in the rod and cap, and machining the sides of the big-ends to provide the specified endplay of the rods on their crankpins, must be done in the correct sequence so the caps will fit correctly and the big-end bores will be round when the rod is finished.

A rod's center to center length, which is the distance between the centers of its piston pin bore and big-end bore, determines how high the piston on the rod will rise in its cylinder at the end of exhaust and compression strokes. This has an effect on compression ratio. The higher the piston head's position in the cylinder at the end of compression strokes, the smaller the volume of the cylinder's combustion chamber will be and the higher the cylinder's compression ratio will be. If the piston head rises to too high a point in the cylinder it will hit the cylinder head. This can cause strange noises and damage the parts involved. To minimize this possibility in engines assembled with minimum deck clearance, which is the clearance between the piston head and the top of the cylinder block when the piston is at top center, at least one manufacturer makes the center to center length of his rods .010-inch shorter than that of standard rods. This difference in length compensates for the amount the rods' shanks expand as the rods become heated to their normal operating temperature. It prevents the rods' center to center length from becoming greater than that of standard rods.

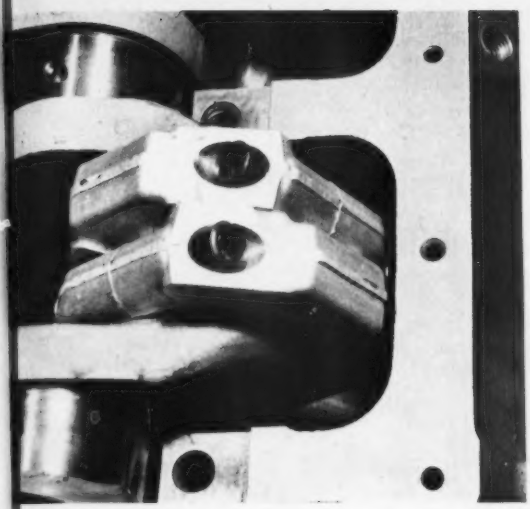
Equal connecting rod weights are essential to correct engine balance. To eliminate or at least minimize problems concerned with rod weight the manufacturers of aluminum rods "selective balance" their rods by selecting and grouping those that have similar weights. In other words, the rods sold as a set will be so nearly identical in weight that they are ready to install. Rods of the same make but sold as parts of different sets quite possibly could not be used interchangeably because there would be too much difference in their weights.

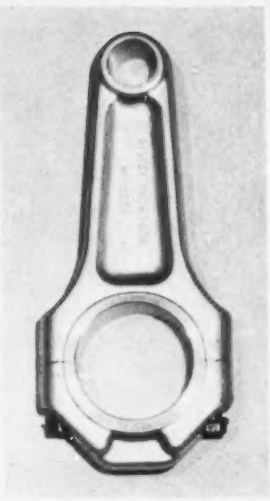
If a fellow decided to balance his rods to closer tolerances than that used by their manufacturer the only material he should remove from them to reduce their weight is the "flash" left on their sides when they are cut from the rough-forged blanks. This is the only excess material on them. Removing material other than this could create weak spots. Rods of some makes are finished by the manufacturer to the extent that the flash is removed by buffing or some other process. No attempt should be made to remove material from buffed rods for balancing purposes. Actually, the quantity of material that must be removed from an aluminum rod to reduce its weight a gram or two makes it rough to reduce the weight for any purpose without changing the rod's contour more than desired.

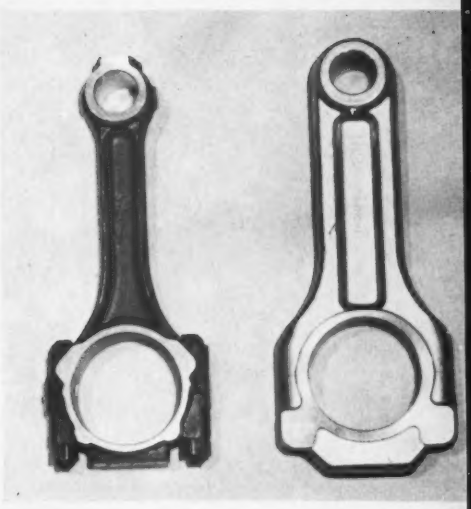
Regardless of how rod weight is established it is important that the engine's crankshaft and rod and piston assemblies be balanced as an assembly before they are installed. This is something that has been explained in another section of this book.

The final proof of any product is the way the product acts in service. As far as this test is concerned aluminum rods have proved themselves to be as represented. They have demonstrated themselves in the wildest of competition engines to be equal to the task for which they were designed by increasing bearing life to many times what it is with standard rods and by not breaking. And these are rods that range from a few grams to a few ounces lighter than those they replaced.

DON FRANCISCO







ABOVE, LEFT—These Howard rods are on a 283 Chevy crankshaft that has been stroked 5/8-in. Block clearance is adequate.

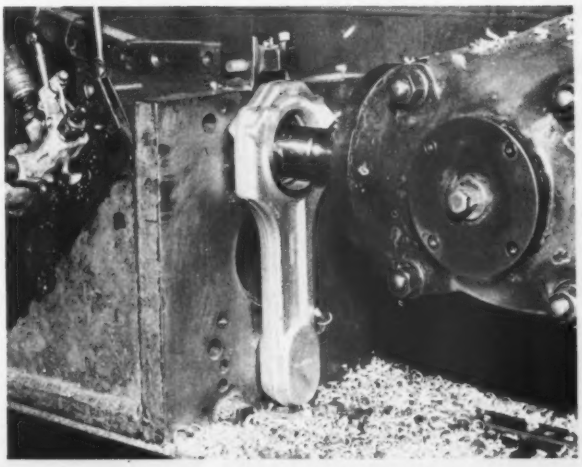
ABOVE, MIDDLE—Mickey Thompson's rod for Chevys has standard cap design. Cap is retained with studs and special nuts.

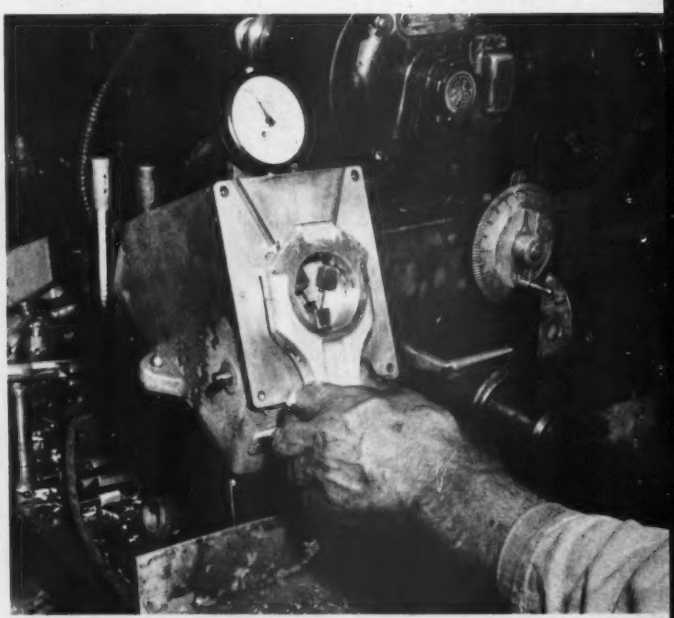
ABOVE, RIGHT—Mickey Thompson forged alum. rod for Pontiac engine and a stock rod. Bulkiness of alum. rod is apparent.

RIGHT—At Howard's Cams, bores of the forged rods are rough-machined on this milling machine fitted with special fixture.

BELOW, LEFT—At Howard's, the rod bores are finished to size with a hone. Honing guarantees round, correctly sized bores.

BELOW, RIGHT—Accurate measuring equipment is necessary for checking rod bores. This dial indicator is used at Howard's.





ROLLER TAPPET CAMSHAFTS

OVERHEAD valve V8 engines modernized the automotive scene for both the general public and hot rodders but they brought with them a problem that even now, after twelve years of research and cut and try experimentation, still hasn't been completely solved. This is the problem of camshaft lobe and lifter wear. Wear of this nature was found only rarely in flathead engines. The apparent reason for it in overhead valve engines is the considerably stronger valve springs required to overcome the greater weight and friction of their more complicated valve actuating mechanism. These stronger springs increase the pressure between tappets and their cams to such a high value that the lubricating oil on their surfaces is squeezed from between them, allowing metal to metal contact between the parts as the cams slide across the tappets. This sliding friction causes rapid wear.

Cam and lifter wear is especially discouraging in reworked engines because it is aggravated by the shape of the cams necessary for high-performance valve action and the even stiffer valve springs the cams require for the high crankshaft speeds competition engines must turn. Grinders of special camshafts have tried every possibility in their efforts to overcome the problem. Among these are camshafts machined in their entirety from steel billets of different alloys, and hard-facing methods which involve coating the cams' lobes with some sort of extremely hard and durable metal.

Billet shafts and hard-facing are definite camshaft improvements. Both extend cam and lifter life enough to be worthwhile and are used by many cam grinders. But although they are improvements, neither is the ultimate answer.

In 1949 a young Southern California cam grinder who prior to this time had specialized in very successful cams for motorcycles decided to apply the knowledge he had gained from Harley-Davidson roller-tappet cams to automobile engines. This was Chet Herbert; the result was the beginning of a complete line of roller-tappet high-performance camshafts Chet grinds for most of the overhead valve V8 engines popular among hot rodders.

Roller tappets, through their ability to roll rather than slide over their cams, eliminate sliding friction in tappet and cam assemblies. Also, the rolling contact area is much easier to lubricate than a sliding area. The result is an almost unlimited service life for the parts.

Chet's decision to grind roller-tappet camshafts for automobiles was not based entirely on improving cam and lifter wear. The roller tappets made it practical to use valve action much more radical than that possible with flat-tappets. Just as the ability of the rollers to roll rather than slide over their cam lobes cut wear, it also allowed the tappets to roll over lobes that lifted the valves faster and higher and lowered them faster. The theory behind the more radical valve action was to improve horsepower output.

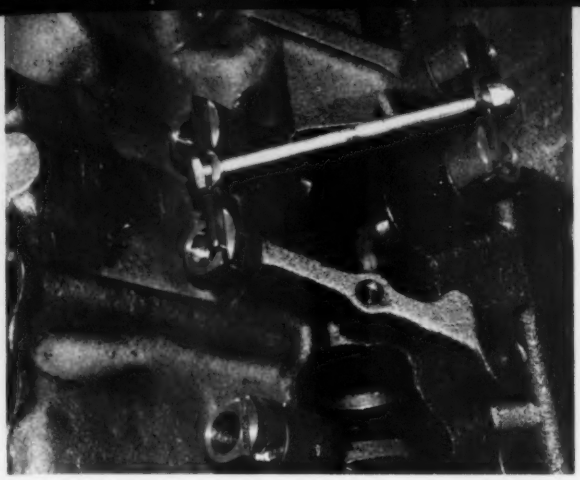
From the cam grinder's standpoint roller tappets have a definite advantage over flat tappets. Their almost hairline contact with the cams at all times simplifies laying-out the valve action they will provide. The contact area between flat tappets and their cams is almost hairline at some points during the valve opening and closing cycle but at others it is much broader. Also, the side of the lobe may strike the tappet's edge and lift it in this manner rather than by pushing against its face. These things complicate the problem of designing a cam that will provide exactly the desired valve action.

Chet's first roller-tappet camshafts were for all-out competition engines only. However, he eventually designed grinds for road engines. These were as successful as the original competition grinds.

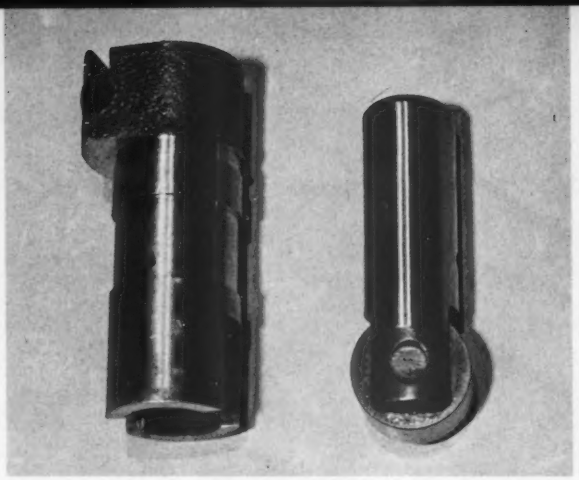
The roller-tappet cam business was all Chet's for several years but then other cam grinders began, one by one, to bring out roller-tappet setups of their own for both competition and road engines. Now, most camshaft companies can supply roller-tappet cams for any engine in which it is possible to install the tappet assemblies. Some engines, such as early model overhead valve Ford V8's, can't be fitted with roller tappets because of the small-diameter tappet bores in their cylinder blocks. The bores must be large enough for the rollers on the tappets to pass through them.

Some early roller-tappet camshafts were noisy because their cams didn't have clearance ramps but nearly all of those now available are just as quiet as flat-tappet grinds. Making the valve action quiet is merely a matter of designing the clearance ramps correctly, as it is with flat-tappet cams.

Competition in the camshaft business has the same effect of stimulating the minds of the men involved as it has in any other business. There wasn't much that could be done to change the camshafts used with roller tappets as the design of their cams requires that a billet shaft be used because the cams on a reground stock shaft wouldn't be large enough, and that for most grinds the shafts be made of steel because



Harman and Collins roller tappets are locked in the cylinder block in pairs with long bolts and eight stamped-steel clips.

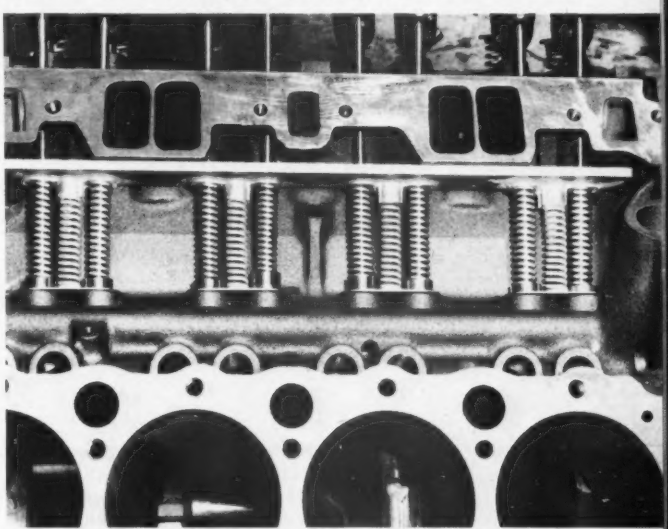


At right is a Harman and Collins roller tappet. It is made of steel. At left is the tappet's guide. It is made of bronze.

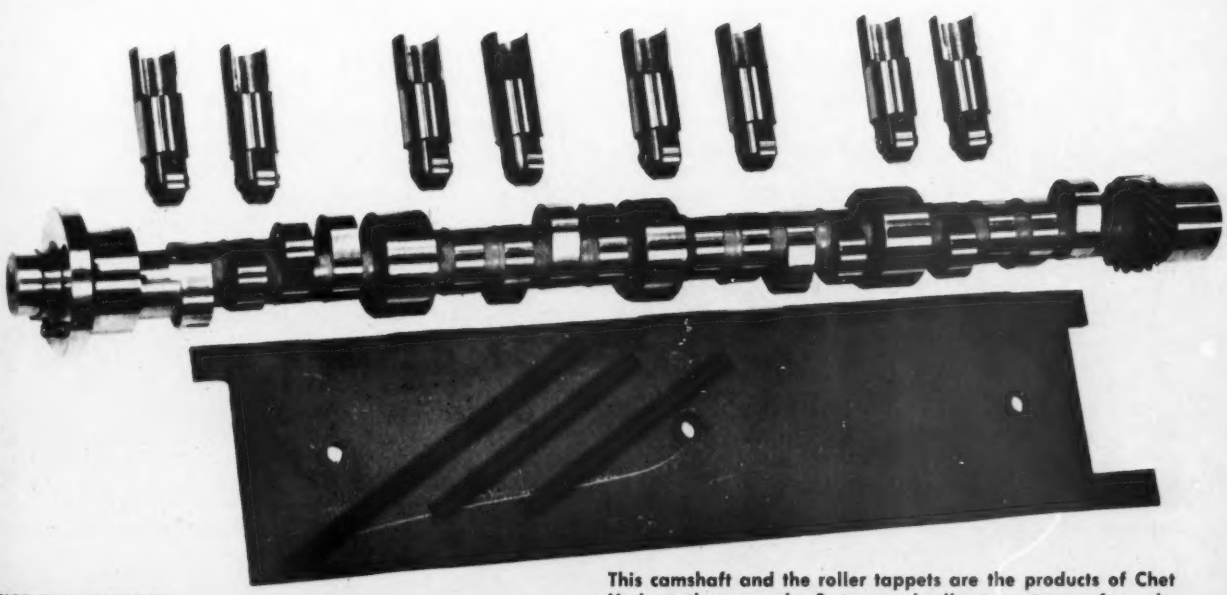
of the loads they must carry. Most grinders use shafts machined from SAE 8620 chrome-moly. The shafts are heat treated before the grinding is finished. At least one grinder has a roller-tappet grind available on a cast-iron shaft.

In contrast to the shafts, the tappets presented all sorts of opportunities for design changes. The most prominent of these is concerned with the way their rollers are held in alignment with the cams. Tappet and cam alignment isn't any problem with the flat tappets for which all modern engines are designed because the tappets are supposed to rotate in their bores in the cylinder block. But when roller tappets are installed, some means must be provided to prevent their rotating so the axis of their rollers will always be parallel with the camshaft's axis. If a tappet should rotate far enough, a side of its roller would be struck by the cam lobe and something would be broken.

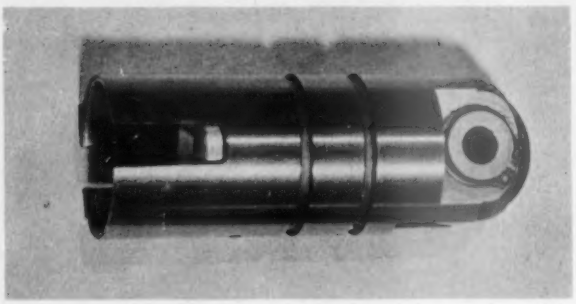
Actually, the rollers are self-aligning except when they are on the heel or lobe of their cams. A considerable amount of pressure would be required to rotate one of them when the valve it controls is opening or closing. This self-aligning



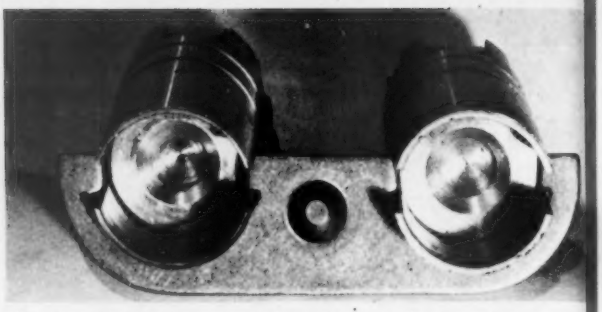
A new roller-tappet setup from Howard's Cams has springs that help hold the tappets on their cams at high crankshaft speeds.



This camshaft and the roller tappets are the products of Chet Herbert, the man who first ground roller tappet cams for rods.



Howard's tappets have slots in their upper ends for the ears on the stamped-steel plates that prevent rotation in the block.



The plate that prevents rotation of Howard's tappets is easy to install and foolproof. A coil spring holds it in position.

tendency is the result of leverage between a roller's face and the face of its cam. When a roller slightly out of line with its cam is contacted by the cam, the pressure of the cam, acting against one side of the roller, rotates the tappet's body on its axis because there isn't any force to resist this movement. But the instant the full width of the roller's face is in contact with the cam, rotation of the tappet body stops because it is resisted by the leverage created by the width of the roller's face acting against the cam's face.

Because of the rollers' self-aligning characteristics, the duties of a tappet aligning device are simple; however, this doesn't reduce the device's importance. It is as necessary as the rollers themselves. The device must be simple as far as installation is concerned and it shouldn't affect the tappets' weight. Installation should be as simple as possible because few rodders have the machine shop facilities and welding equipment they might require for a more difficult installation.

Some of the early tappet assemblies were quite difficult to install the first time but subsequent installations were much simpler after the required modifications had been made to the cylinder block. Now, most cam grinders have what are called "drop-in" lifters that don't require any modifications at all to the cylinder block. These lifters have their own devices for keeping them in alignment. One popular type has an extra-long body that extends above the upper end of the block's tappet bore. Attached to this part of the body is a stamped steel member that acts as a guide for a mating member on the tappet next to it. When the tappets move up and down in relation to each other one guide moves in the other. In this way the tappets for the valves for each of the cylinders act together to hold each other in alignment.

A more recent drop-in tappet uses the cams on the camshaft for aligning purposes. The cams are approximately the same width as the rollers so that extensions of the sides of the lower ends of the tappet bodies can straddle them. These extensions are usually called "struts." A variation of this method, which cuts tappet weight a tiny amount, is to use just one strut. Tappets that have a single strut are installed so the strut is ahead of their cam. This lets the struts serve a second purpose by preventing forward movement of the camshaft in the cylinder block. This, to date, is probably the best tappet aligning method because it adds the least weight to the tappets.

Tappets that depend on their cams for alignment require an accurately machined camshaft. The sides of the shaft's cams must be aligned correctly with the tappet bores in the cylinder block so they will be located correctly in relation to the tappets' struts. Some clearance is allowed between the struts and the cams so that when the rollers and cams are correctly aligned the struts won't exert any pressure on the cams.

Tappet weight is important in any high-performance engine because a heavier tappet requires greater spring pressure to hold it in contact with the lobe of its cam than would a lighter tappet. One of the features of roller tappets that have made them so popular is their ability to successfully withstand the high valve spring pressures necessary for high crankshaft speeds but this doesn't mean that pressures shouldn't be kept as low as possible. The less pressure the springs exert on their lifters, the less total pressure there is from all the springs to push the camshaft out of alignment, the easier it will be to rotate the shaft, and the longer the valve actuating parts will last.

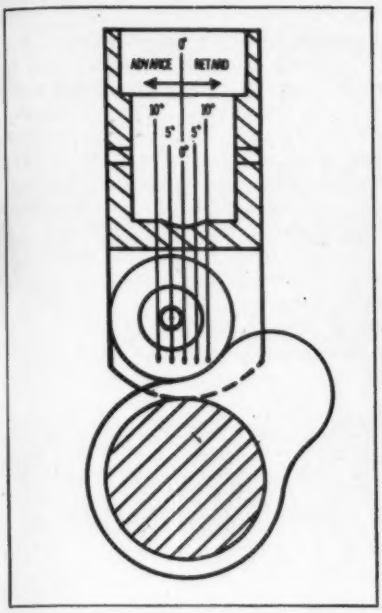
Some grinders of roller-tappet camshafts have recently made available spring setups that help compensate for tappet weight. These consist of coil springs of the compression type that are installed between the upper ends of the tappets and a special plate that rests against the upper surface of the tappet chamber. The plate has recesses that locate the upper ends of the springs in relation to the pushrod openings in the head or block. Pushrods pass through the centers of the springs.

The purpose of these springs is to help hold the lifters in constant contact with the cams. The pressure they exert isn't great—those in Iskenderian's "Ultra-Rev" kit have a maximum pressure of 30 pounds—but it is said to be enough to effect an appreciable increase in an engine's maximum speed before valve float occurs.

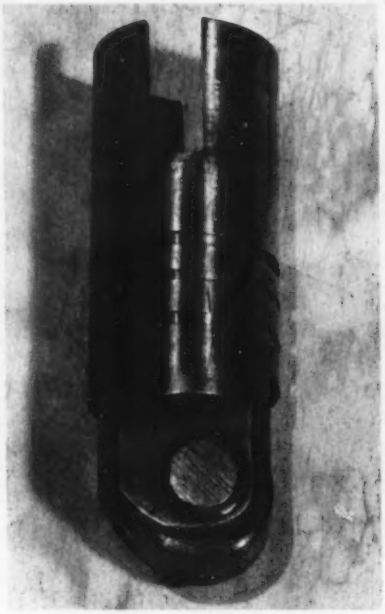
Most tappets have needle bearings for their rollers but some have plain bushings. The purpose of the bearing is to minimize friction between the roller and its support when the roller rotates. It must be strong enough to carry the valve spring load without galling or breaking.

A recent addition to the roller-tappet camshaft field is the shaft ground by Bus Schaller. Schaller's shop is in Turlock, Calif. His roller-tappet camshaft differs from all other camshafts of both standard and special types by having cams that have two lobes rather than one for each of the engine's valves so it can be rotated at one-quarter rather than the conventional one-half crankshaft speed.

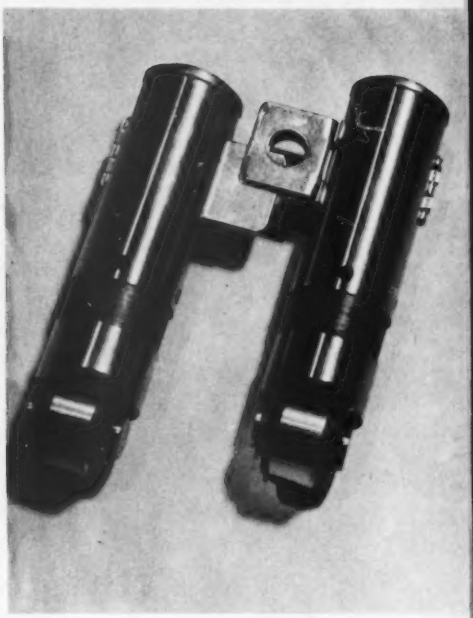
Actually, Schaller's quarter-speed camshaft is the first major design change that has been made in automotive engines since internal combustion engines first became practical for automotive applications. What Schaller originally had in mind when he started work on it was a method of reducing friction in the engine. This would automatically increase the power delivered by the engine's flywheel. His theory was that a camshaft that rotated at one-quarter crankshaft speed would rotate easier in its bearings than one that rotates at one-half crankshaft speed. Also, the greater leverage between the crankshaft and the camshaft, made possible by the one to four ratio, would reduce the amount



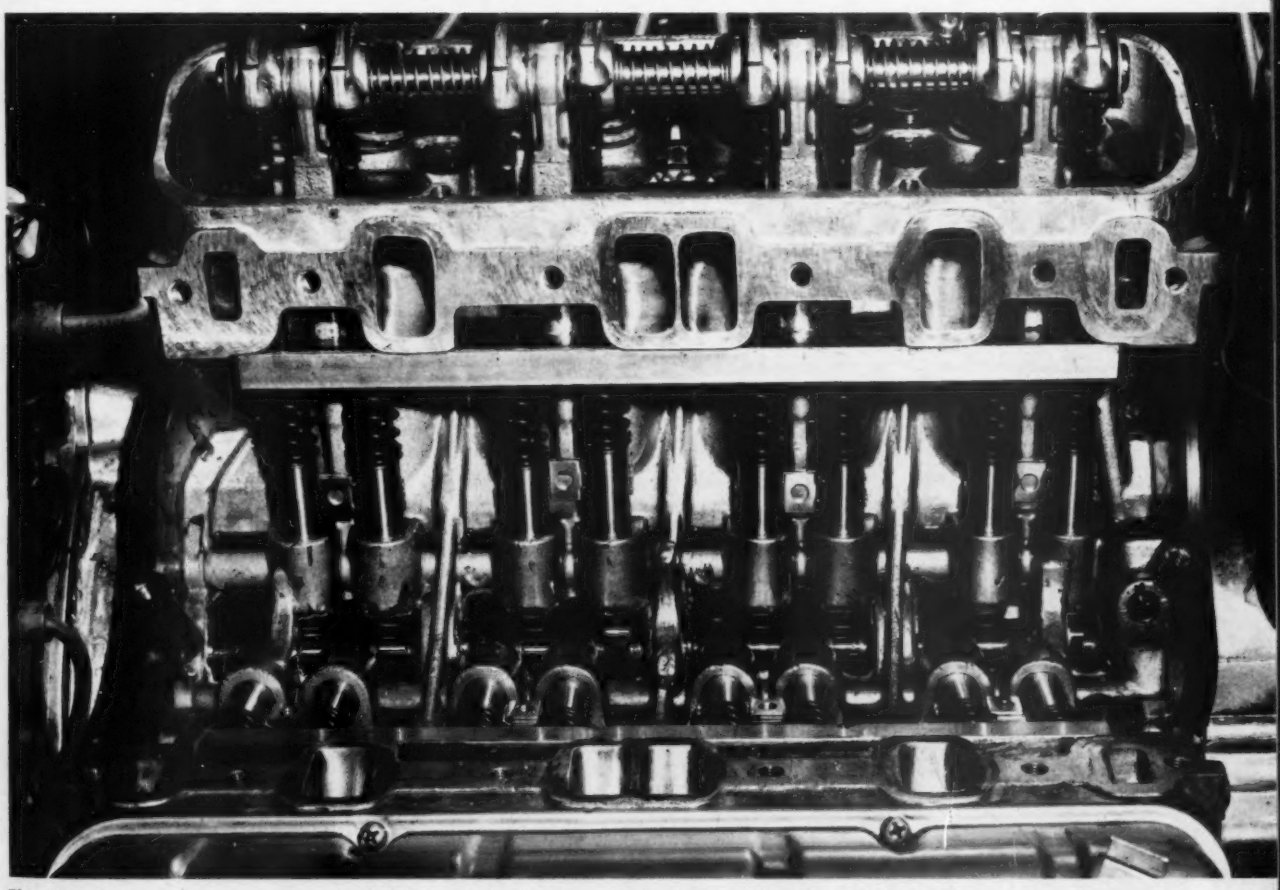
Howard has a special roller tappet that has an offset roller. Reversing the tappet in the block changes the valve timing.



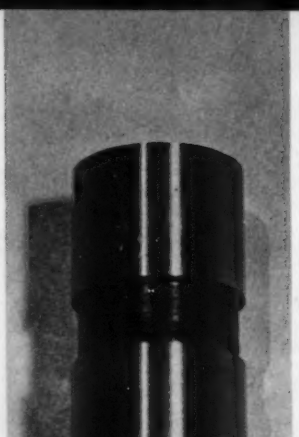
Howard's reversible tappets will be made with different amounts of roller offset to vary the amount they change the timing.



Iskenderian roller tappets must be extra-long because of the device used to keep them from rotating in the cylinder block.

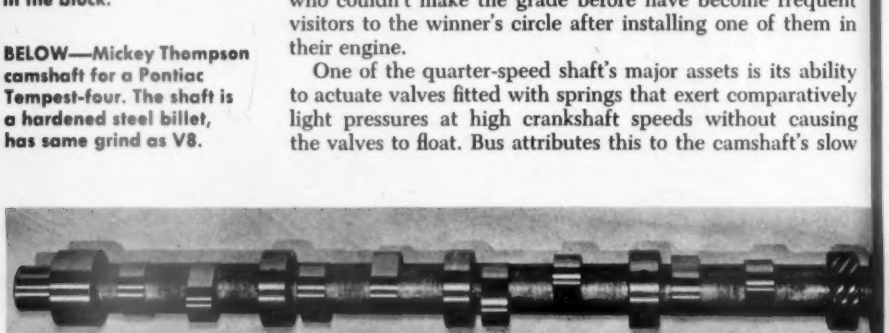


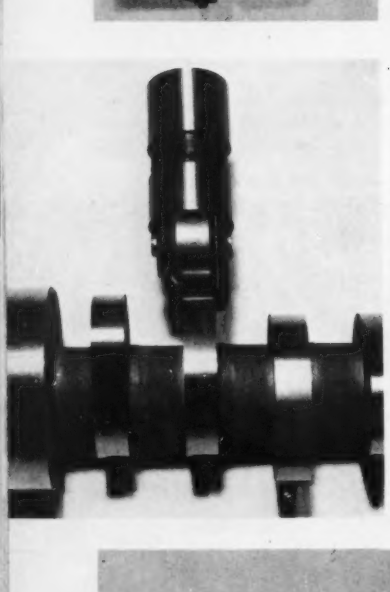
The Iskenderian roller-tappet setup in this little Buick aluminum V8 has Isky's "Ultra-Rev" kit that adds springs to the tappets.



LEFT—Drop-in tappets from Frank McGurk Engineering have a single strut that prevents them from rotating in the block.

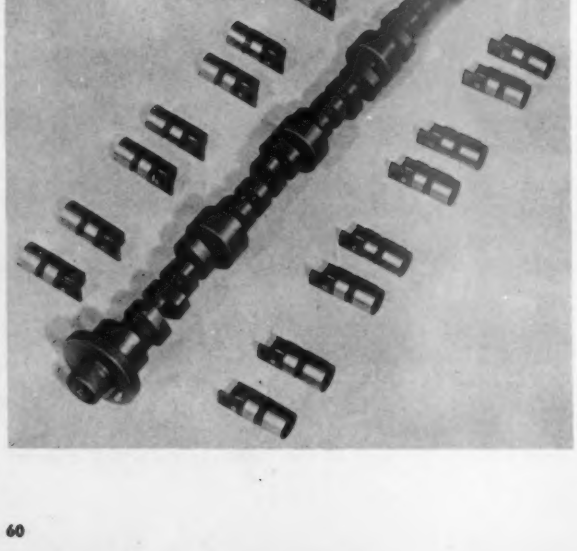
BELOW—Mickey Thompson camshaft for a Pontiac Tempest-four. The shaft is a hardened steel billet, has same grind as V8.





LEFT—McGurk tappet and cam assembly, showing how the strut aligns with the cams. Spacing of the cams on the shaft is important.

BELOW—Roller tappets and camshaft by Dempsey Wilson also use single-strut. Dempsey has at least one grind on cast-iron billet.



of crankshaft torque required to rotate the camshaft.

Schaller's theories have been discussed both pro and con by authorities and would-be authorities in the mechanical engineering field but as yet no one has definitely proved them to be either correct or incorrect. However, the theories have become secondary in importance to the actual results obtained with the camshaft. Many competition car owners who couldn't make the grade before have become frequent visitors to the winner's circle after installing one of them in their engine.

One of the quarter-speed shaft's major assets is its ability to actuate valves fitted with springs that exert comparatively light pressures at high crankshaft speeds without causing the valves to float. Bus attributes this to the camshaft's slow speed of rotation. Because it is rotating only one-half as fast as a conventional half-speed camshaft at any specific engine speed, the lobes on the shaft's cams act more gently against the tappets although the valves are opened and closed at the same rates as with a half-speed shaft. This slower, gentler action allows the tappets to remain in contact with the cam lobes during the complete opening and closing cycle. The more severe action of the faster moving lobes on a half-speed shaft's cams have a tendency to knock the lifters away from them. This is the reason such strong valve springs are required with half-speed cams. The purpose of the greater spring tension is to hold the tappets in contact with the cams at all times. Just how efficiently they do this is food for debate.

The gentle action of Schaller's camshaft is also thought to be responsible for at least a part of the better engine performance it provides over that obtained with flat-tappet cams that have similar valve timing specifications, and with which the same valve springs are used. The theory is that by eliminating any tendency toward valve float the valve timing remains the same at all engine speeds. This is in contrast to the variations that occur in the timing with a half-speed shaft when the rough action of its cams begins to throw the tappets rather than lift them. This can occur at quite low crankshaft speeds.

Another feature of the quarter-speed shaft that undoubtedly has much to do with its ability to actuate valves at high engine speeds without causing them to float is the shape of its cams. A characteristic of the cams is that the more radical they become to make the valve timing longer for efficient breathing at high crankshaft speeds, the closer to round they become. This characteristic isn't shared by the cams on half-speed shafts. As the valve action provided by half-speed cams becomes more radical, the shape of the cams comes closer to that of a rectangle that has rounded ends. It's easy to visualize how a tappet can follow a cam that has a basically round shape easier than it can one that has sides that are nearly flat and that is rotating twice as fast.

Schaller eliminated a weak link in the valve actuating mechanism by replacing the sprocket and chain setup that

drives the camshaft in most automotive engines with a pair of gears. When chain and sprocket assemblies are used with special camshafts they have short service lives. The chains stretch and the sprockets wear. This allows the valve timing to become retarded in relation to the positions of the pistons in the engine's cylinders. The hardened-steel crankshaft gear in Schaller's drive setup rotates a large duralumin camshaft gear. The design of the gears and the facts that the one for the camshaft has four times as many teeth as the one for the crankshaft and that the quarter-speed shaft rotates so much slower and also easier because of the much lighter valve spring pressure gives the gears a potential service life equal to that of any moving part in the engine.

A durable drive mechanism for the quarter-speed camshaft and accurate placement of its cams in relation to the shaft and to each other are extremely important. The effects slack in the drive mechanism or errors in the cams' placement would have on valve timing would be magnified four times rather than the two times for conventional half-speed shafts. This is the result of rotating the camshaft at one-quarter rather than at one-half crankshaft speed. One degree of its rotation is equal to four degrees of crankshaft rotation whereas one degree of a half-speed shaft's rotation is equal to only two degrees of crankshaft rotation.

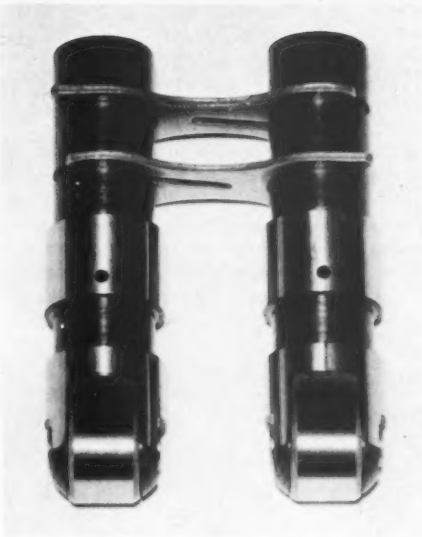
The roller tappets Schaller is using at the present time are of the drop-in type. They have two struts that straddle the camshaft's lobes. Stock pushrods and rocker arms are used.

Reducing the camshaft's speed of rotation necessitated changes in some of the engine's other components. A new front cover, or timing gear housing, had to be made for the cylinder block to provide room for the larger cam gear. This consists of an aluminum housing and a separate aluminum cover.

Driving the camshaft with gears rather than with sprockets and a chain reversed its direction of rotation. This, and the shaft's lower speed of rotation, made necessary a means of returning the direction and speed of rotation of the distributor and oil pump shafts, which are driven by the camshaft, to normal. The direction of rotation could have been reversed by merely reversing the angles of the drive and driven teeth on the camshaft and distributor shaft gears but there would still have been the problem of boosting the speed of rotation. Both problems were solved by inserting a set of planetary gears between the camshatf and a standard Corvette distributor shaft that rotates the distributor's breaker cam and the oil pump's shaft.

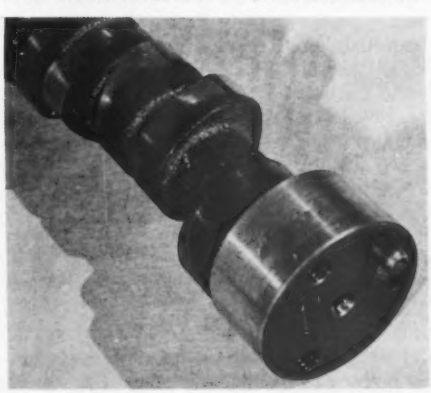
The planetary gears, which are mounted in an aluminum housing that occupies the standard distributor opening in the cylinder block, consist of a ring gear, four planetary gears, and a sun gear. When the ring gear is rotated by the shaft that passes through the aluminum housing and has a gear on its lower end that meshes with the drive gear on the camshaft, it rotates the planetary gears, which are held in place by steel pins. The planetary gears, in turn, rotate the sun gear, which is connected to the standard distributor shaft in the assembly. Standard Corvette advance weights, breaker points, condenser, rotor, cap, etc., are used in the distributor. Breaker point action is improved over that provided by a standard Corvette distributor by a large ball bearing directly under the governor weight plate on the distributor's shaft. This bearing holds the shaft and the breaker cam in their correct positions at all crankshaft speeds.

Materials used for the camshaft and the parts required for its installation are the best available. The shaft itself is ground from a 4140 chrome-moly billet. It is heat treated to



Bruce Crower's tappets have extra-wide rollers. The tappet bodies are long to provide bearing area for something new in anti-rotation clips.

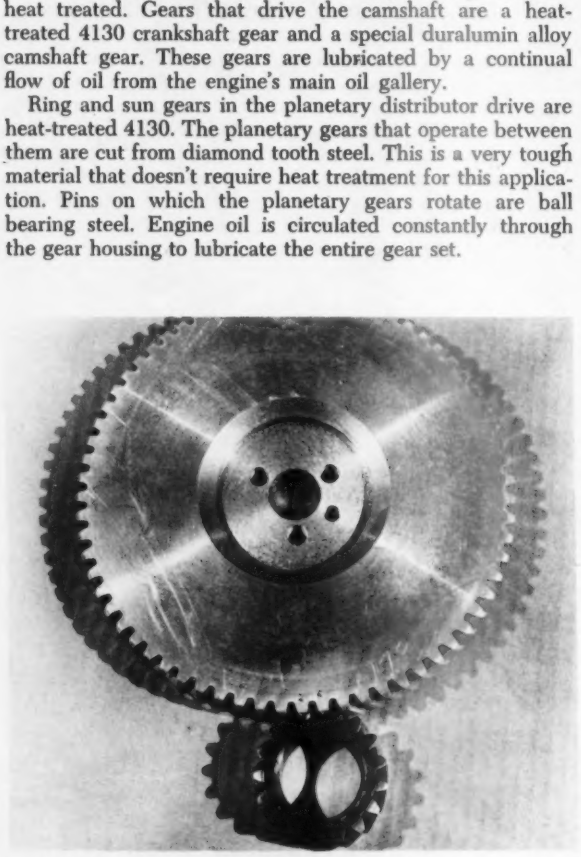
The cams on Bus Schaller's highly successful new double-lobe camshaft differ from those on other shafts because the shaft rotates slower.



Valve timing provided by this Chevrolet Duntov camshaft is the same as that of the particular Schaller double-lobe ¼-speed shaft above.

give it maximum wear resistance. The drive gear for the distributor assembly is 4130 chrome-moly. This gear is also heat treated. Gears that drive the camshaft are a heat-treated 4130 crankshaft gear and a special duralumin alloy camshaft gear. These gears are lubricated by a continual flow of oil from the engine's main oil gallery.

Ring and sun gears in the planetary distributor drive are heat-treated 4130. The planetary gears that operate between them are cut from diamond tooth steel. This is a very tough material that doesn't require heat treatment for this application. Pins on which the planetary gears rotate are ball bearing steel. Engine oil is circulated constantly through the gear housing to lubricate the entire gear set.

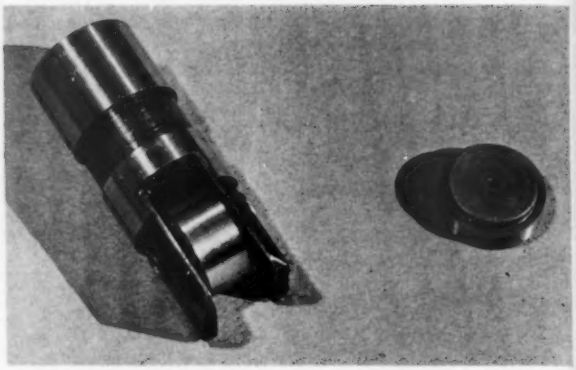


Schaller's camshaft is driven with gears instead of a chain and sprockets. Their ratio is one to four rather than one to two.

The camshaft and other necessary parts are sold in kit form. Their installation is merely a matter of installing them in place of the standard parts they replace. No special machine work of any sort is required by the engine or the special parts.

When camshaft grinders first began making roller tappet camshafts for Chevrolet V8's they had quite a bit of trouble designing the shafts and tappets so they would meter the oil supply to the rocker arm and valve assemblies correctly. Most assemblies would pass so much oil that the rocker arm chambers would literally become flooded. This allowed excessive quantities of oil to flow down the valve stems and into the combustion chambers.

The oil supply for Chevy V8 tappets and the rest of the valve actuating mechanism originates at the engine's rear camshaft bearing bore. Oil from the main gallery enters the bore through an inlet port in its side and leaves through two outlet ports. However, oil can flow from the inlet port to an outlet port only when the two are connected by a passage that is nothing more than a flat spot milled on the camshaft's rear bearing journal. This flat spot connects the ports during a few degrees of camshaft rotation each time the camshaft makes a full revolution. Its length determines the amount of oil that can flow to the outlet ports. This is the first point in the system where the oil is metered.

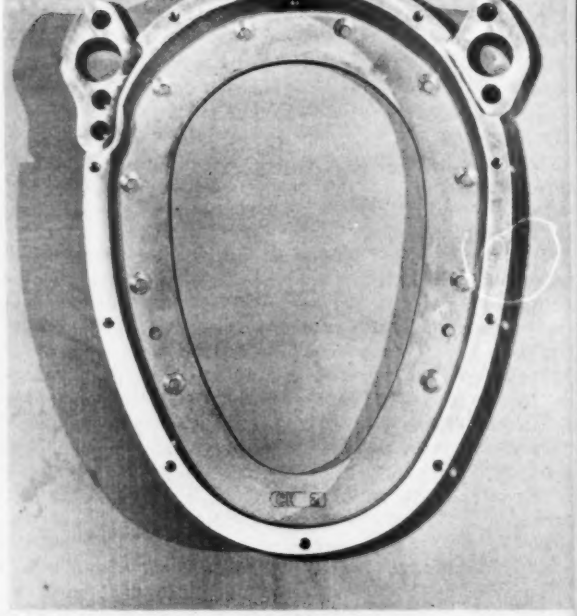


The tappets Schaller uses have two struts to align them on the camshaft. Racer Brown uses the same tappets for his camshafts.

From the outlet ports the oil flows into a pair of galleries that run the full length of the cylinder block. These galleries intersect the block's tappet bores. Their main purpose is to deliver oil to the tappets. Oil enters the tappets, which are hollow, by means of an opening drilled through their side. The size and position of this opening are critical as it is the second point in the system where oil flow is metered.

Oil that enters the tappets flows upward through openings in their pushrod seats and into the hollow pushrods. It continues up the pushrods to the pushrod seats in the rocker arms and through a hole in the arms. Part of it then flows into the fulcrum ball seat in the arms to lubricate the seats and some of it is splashed onto the valve ends of the arms and the valve and spring assemblies by normal rocker arm motion.

There is another way of lubricating the valve action that



The large camshaft gear for Schaller's camshaft requires this oversize housing. It is cast-aluminum and bolts to the block.

doesn't depend on the cams and lifters and that allows the engine's owner to easily adjust the amount of oil that flows to the rocker arms. This is with a "drip system" like the one Tom Cobbs built for his 283 Chevy. Tom is well-known in hot rod circles for the engines and cars he has built. He was one of the first rodders to get really satisfactory results from blowers. He designed his drip system for an engine he was using for a series of dynamometer tests with roller-tappet camshafts before some cam grinders had solved the rocker arm lubrication problem.

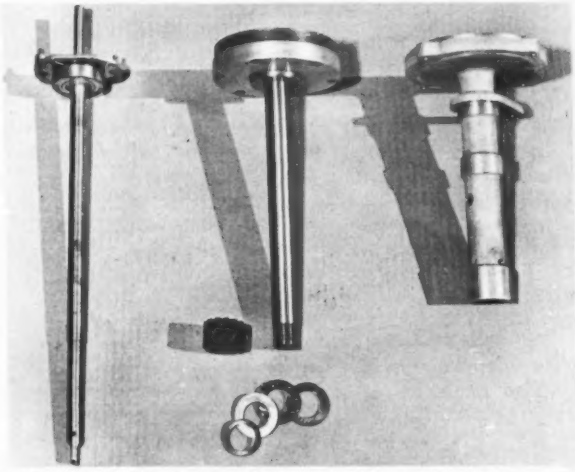
Although the drip system was practically a necessity when Tom built it, camshaft manufacturers have since made it almost unnecessary by bringing the rocker arm oiling situation under much better control; however, on some types of competition engines it is still a highly desirable feature because of the complete control it provides over the oil delivered to the valve assemblies and the freedom it gives from camshaft and tappet considerations.

The drip system is really quite simple. It consists of a method of taking oil from the engine's main oil gallery and allowing it to literally drip into the rocker arm sockets. Normal motion of the arms throws oil onto their pushrod ends, where it flows through the standard feed hole to lubricate the upper ends of the rods, and onto the valve ends of the arms and the valve stems.

Tom made the system in two branches. Each branch has a small manifold to which the necessary hoses and the valves that regulate the oil flow are connected.

The manifolds were made from 1-inch square aluminum. A passage $1\frac{1}{32}$-inch in diameter, which is the diameter for a ⅛-inch pipe thread, was drilled lengthwise through each of them. Four additional $1\frac{1}{32}$-inch passages were drilled in the manifold for the right cylinder bank and two were drilled in the one for the left bank. All the passages were positioned to form branches of the lengthwise passage. Ends

To return the direction and speed of the distributor and oil pump shafts to normal, Bus uses this special planetary gearset.

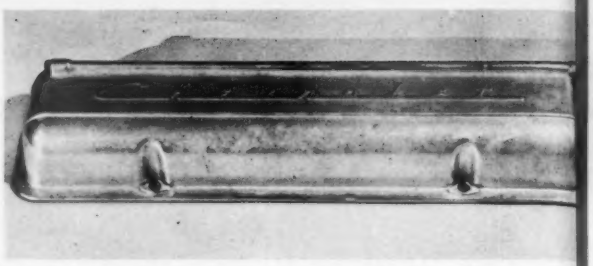


Parts for Schaller's distributor are, left to right, shaft and governor plate assembly, driveshaft for the planetary gearset with its thrust washer and gear, and the driveshaft's housing.

The cover for Schaller's special gear housing is also aluminum. The housing and cover illustrated are for small Chevrolet V8's.

This is the special distributor setup that must be used with Schaller's cam.

After the oil leaves holes in the top of the rocker arm cover and the distributing tube, gravity moves it to the arms.



The thin-wall steel distributing tube was soft-soldered to the top of the rocker arm cover and the cover was cadmium plated.

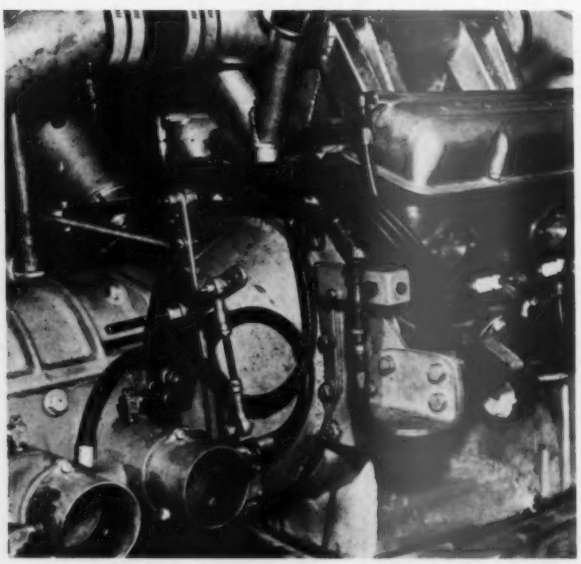
of all the passages were threaded with a ⅛-inch pipe tap.

Three of the passages in the right manifold were drilled in the surface that faces forward when the manifold is bolted to the front surface of its respective cylinder bank. One of these supplies oil to the engine's GMC blower—it wouldn't be necessary if the manifold were for rocker arm lubrication only. Another is the manifold's oil inlet. A hose from a threaded opening in the front end of the engine's main oil gallery is connected to it. The opening in the gallery is a standard feature of these engines. It is plugged with a ⅛-inch pipe plug. All Tom had to do was remove the plug and replace it with a fitting of the correct type. The third passage is for the hose to the right rocker arm cover.

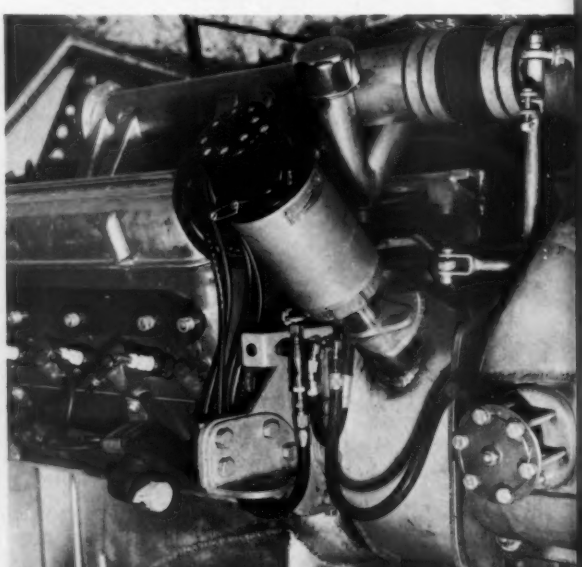
The fourth passage in the right manifold is in the manifold's upper surface. A hose connects it to a corresponding passage in the upper side of the left manifold. Oil that enters the right manifold's inlet flows through this hose to the left manifold. The second passage in the left manifold is in its forward side. This is the outlet to the rocker arm cover.

The rocker arm covers were adapted to the drip system by drilling eight 3/16-inch holes in the top of each of them directly over the fulcrum ball sockets in the rocker arms. Thin-wall steel tubes that have a ¼-inch inside diameter and to which Tom fitted ends that have ⅛-inch pipe threads were soldered to the covers over the holes. Openings drilled in the tubes with a number 58 drill were aligned with the holes in the covers. The rear end of each tube is sealed with a pipe plug and the forward end is fitted with an elbow to which the oil feed line from the drip system manifold on the cover's side of the engine is connected. These hoses are long enough to enable the covers to be removed without disconnecting them.

Oil that leaves each manifold and enters the hose to the rocker arm cover flows through a "sight gauge." It can be seen through a port in the side of the gauge's metal housing

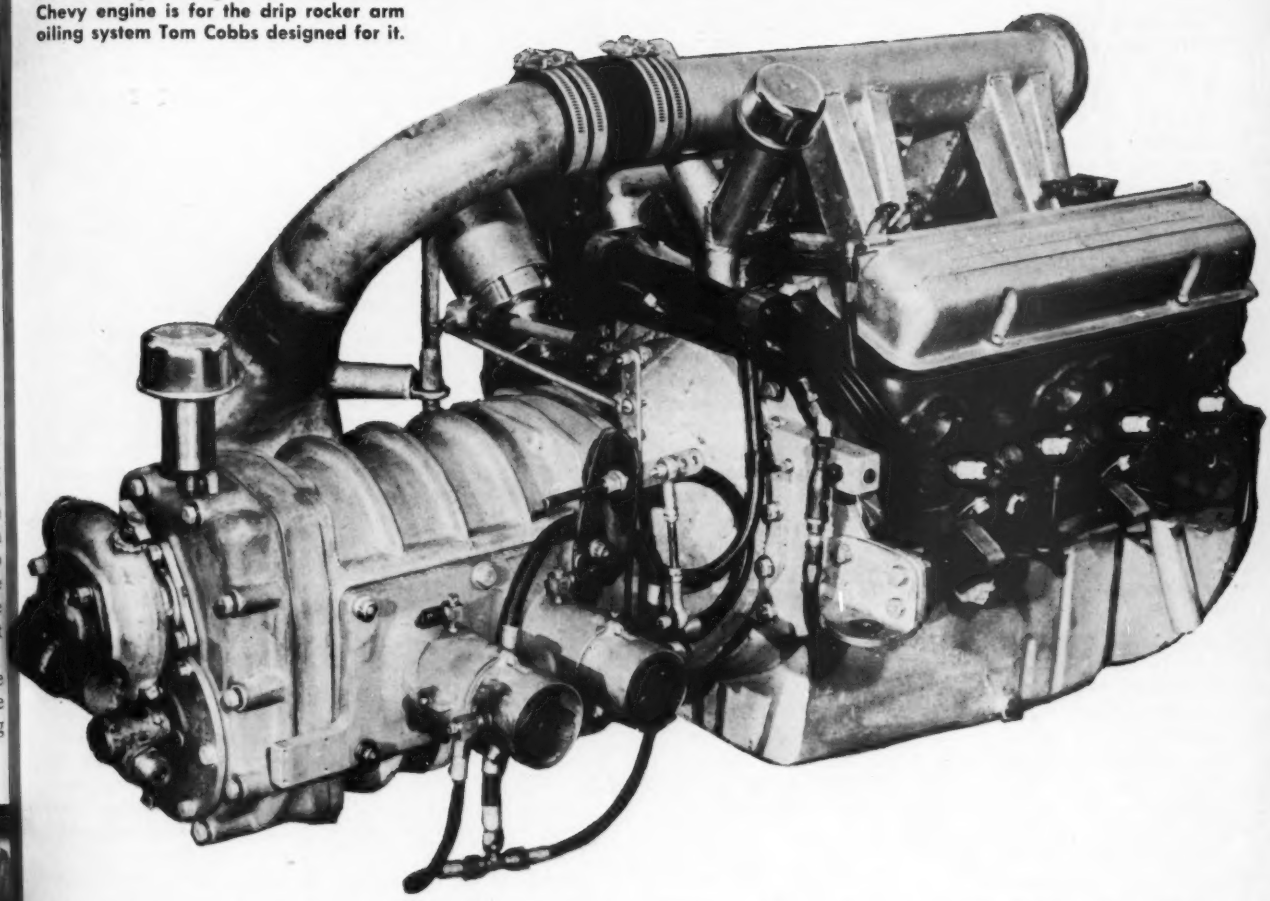


ABOVE—Oil flow to the rocker arms through the drip oiling system Tom Cobbs built for his Chevy engine is controlled by the "sight" gauge just above the motormount pad.



RIGHT—Oil flow to the rocker arms on the right side of the engine is controlled by a sight gauge. Aluminum manifolds serve as terminal points for the gauges and hoses.

Part of the plumbing on the front of this Chevy engine is for the drip rocker arm oiling system Tom Cobbs designed for it.



and the wall of a glass tube inside the gauge. Its rate of flow is changed by screwing an adjusting needle in or out of the gauge. Such a small quantity of oil is required that the adjustment need be only approximately one turn open. The amount of oil that flows through the gauges varies with the oil's viscosity rating and its temperature. However, temperature doesn't pose any problem because once the gauges have been adjusted for the oil being used the rocker arms receive adequate lubrication when the oil is either cold or hot.

It would be possible to build a drip system without the sight gauges by using orifices approximately 1/16-inch in diameter in the lines between the oil distributing manifolds and the rocker arm covers. This could be done quite easily by filling one of the fittings for each line with soft solder and then drilling a hole of the desired diameter through the solder. Some experimentation might be required to determine the exact orifice diameter that would pass the correct amount of oil but this shouldn't prove to be difficult. The lower the oil's viscosity, the smaller the orifice would have to be.

In addition to permitting precise regulation of the quantity of oil deliverd to the rocker arms, the drip system makes it possible to operate the engine with empty pushrods rather than pushrods that are partially or completely filled with oil. Keeping the oil out of the pushrods effects some reduction in their weight. However, Tom believes that a factor possibly more important than the weight reduction is that there isn't any oil in the rods to surge back and forth and create inertia forces when the engine is running at high crankshaft speeds. These forces could possibly prevent the rods from stopping as quickly as they should when the tappets lift them to open the valves. When the rods are empty, there isn't any possibility of such action.

To prevent oil from entering stock pushrods used with a drip system, the openings in their ends must be plugged. This can be done with soft solder. Use plenty of soldering flux to clean the surfaces of the openings so that the solder can bond itself tightly to the metal. When buying special pushrods be sure they do not have openings in their ends.

Installation of a drip system doesn't require any changes to the normal oil supply to the engine's valve lifters if the ends of the pushrods are plugged. Lifters of any type may be used in their stock form.

According to Tom, the main precaution to observe with a drip system is to be sure the oil in the engine isn't allowed to become dirty. It's possible that dirty oil might plug the small openings in the cover tubes and prevent oil from reaching one or more of the rocker arms. It is easy to clean the passages through the manifolds and the cover tubes by running a rifle barrel brush of the correct size through them but it's better to keep the oil clean so that dirt doesn't collect in the passages. This isn't any problem with engines that have full-flow oil filters. **DON FRANCISCO**

HOT IGNITIONS

ONE OF an automobile engine's standard components that can do its job perfectly on a stock engine but that will fall flat on its face after the engine has been reworked is its ignition system. Compression pressures in stock engines are low compared to those in either naturally aspirated or blown reworked engines and the valve action of stock engines acts as a governor to keep crankshaft speeds comparatively low. After compression pressures have been raised by various reworking procedures and the valve action improved to raise an engine's maximum crankshaft speed to almost double what it was previously, only a highly efficient ignition system can provide positive firing of the fuel and air mixture in the engine's cylinders.

The two basic types of ignition systems are battery and magneto. Battery systems are standard on all automobile engines built in the United States. Reasons for this are their lower first cost and simpler maintenance procedures. It is possible to manufacture a battery ignition distributor, which is the heart of the ignition system, of much cheaper materials and to much sloppier tolerances than would be acceptable in a magneto. These low-grade materials and poor workmanship can do the job in a stock engine but not in one that has a lot of compression pressure and rpm's.

Battery and magneto ignition systems are quite similar in their operation. The difference between them is that a battery system requires a battery of some sort as its source of primary current, hence its name, whereas a magneto generates its own primary current. This is another reason battery systems are so popular for automobiles. As long as a car requires a battery for its starting motor and lights it makes good sense to also use the battery to operate the ignition system, especially when the battery system is capable of doing the job within the engine's limitations.

In some types of competition cars a battery is just unnecessary weight and trouble. As a magneto will provide ignition as good or better than that possible with a battery system, most fellows are happy to install a mag and forget about the battery. This doesn't mean that magnetos aren't used in the engines of cars that also have batteries. Far from it. Plenty of drag cars that travel to and from the drag strip under their own power and carry their owners to and from work during the week are running on a magneto.

An ignition system functions on both electrical and mechanical principles. Electrically, it converts the low-voltage primary current of a battery or the primary current created in the low-voltage portion of a magneto to a secondary current of much higher voltage. The actual voltage of the secondary current is determined by the resistance between the electrodes in the engine's spark plugs. Mechanically, the system distributes the secondary current to the plugs in the correct sequence to match the engine's firing order, and at the correct times as determined by the positions of the pistons in the cylinders.

The voltage of the battery in a battery system or that generated by the low-voltage portion of a magneto is converted to secondary voltage by running it through an induction coil. When current from either of these sources is flowing through a coil's primary winding it creates a magnetic field around the coil. When the current flow is stopped, by opening a set of breaker points in its circuit, the magnetic field collapses and induces voltage in the coil's secondary winding. The secondary voltage is then conducted to a rotor which provides a path for it to one of the terminals in a cap or terminal plate of some type. Each of the terminals is connected to a spark plug in one of the engine's cylinders by means of a suitable cable. As the engine's crankshaft rotates, the rotor moves from one terminal to another.

Where a battery system fails on a reworked engine is in its ability to create a magnetic field of sufficient strength around the induction coil to enable the coil to deliver the necessary secondary voltage to overcome the resistance between the electrodes in the spark plugs. This failure is due mainly to the lack of time available between secondary discharges at high crankshaft speeds to allow sufficient flow of primary current to create a large enough field. Another thing that can cause trouble in this respect is faulty action of the breaker points in the primary circuit. Points in standard distributors aren't designed to open and close as rapidly as they must at competition engine speeds. They will sometimes fail to close as quickly as they should and sometimes they will bounce open again after they have closed. Either of these actions shortens the time during which primary current can flow through the coil. They have a disastrous effect on an engine's performance.

Primary current flowing through the induction coil of a magneto differs from the primary current in a battery system by being strongest at high engine speeds and weakest at low speeds. By gaining in strength as engine speed increases, the current can continue to create a magnetic field of sufficient size around the mag's coil to enable the coil to develop adequate secondary voltage regardless of how high the speed becomes. Satisfactory flow of primary current in a magneto is dependent on good breaker point action just as it is in a battery system, but the points in mags for competition engines are designed to close correctly and remain closed at high crankshaft speeds.

A magneto must be made of the best available materials and to close tolerances.

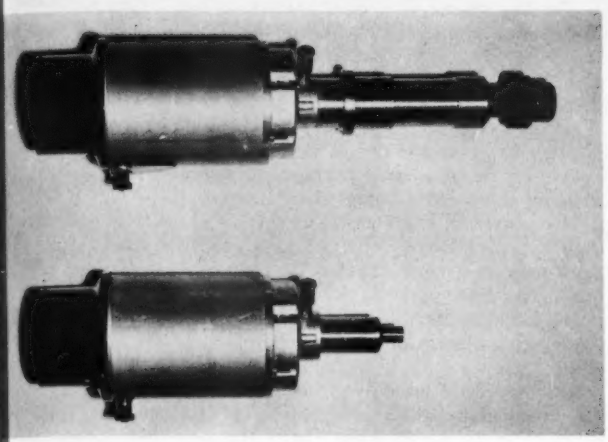
Special ignition distributors for battery systems that have materials and tolerances equal to those found in magnetos are available for car owners who want the best possible battery system. The design of these distributors makes it possible for them to function both mechanically and electrically at any crankshaft speed a reworked engine is capable of turning; however, the system needs a battery.

It's the design of the special distributors that makes it possible for the electrical part of the system of which they are a part to function at high engine speeds. The design differs from that of standard distributors by having two sets of breaker points instead of one and a four-lobe breaker cam instead of an eight-lobe cam. Using a four-lobe cam makes it possible to use two coils in the system.

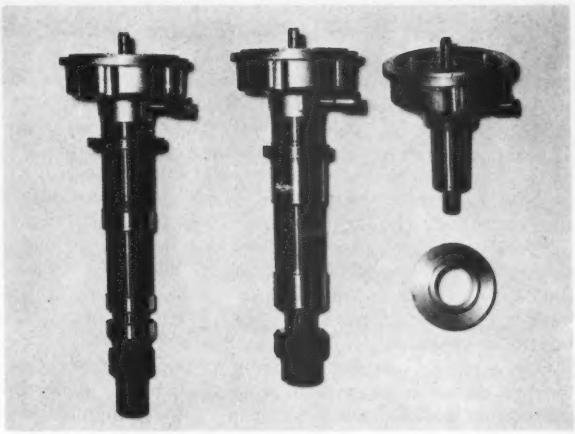
The principle on which two-coil battery ignition systems are based is that of using two four-cylinder systems instead of one eight-cylinder system to fire the engine's eight cylinders. Both systems are contained in the same housing to simplify construction and timing problems.

The effect of using two four-cylinder ignition systems for eight cylinders is to double the period of time each system's breaker points are closed between firing impulses. This, of course, doubles the time during which primary current can flow through each coil to create a magnetic field. Because the points open only half as often, it is possible to design the lobes on the cam that opens them so the rubbing block on their movable arm will follow the lobes at high engine speeds. This eliminates point floating and bouncing, guaranteeing proper coil action.

An engine's power output can suffer badly if the voltage discharges delivered to its spark plugs aren't timed correctly with the pistons in its cylinders. This wouldn't be a



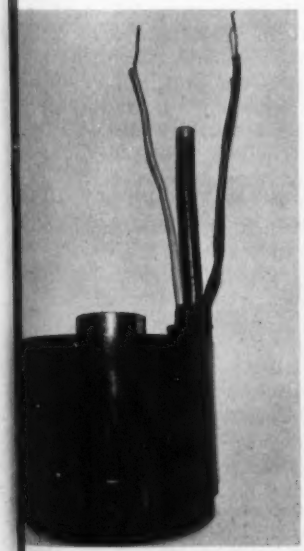
A pair of Scintilla Vertex magnetos. One on the bottom has its stock form, the other has been adapted to fit a V8 engine.

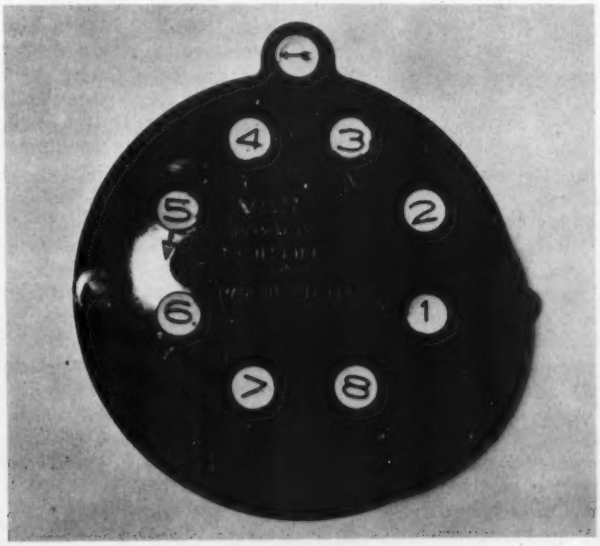


Three different forms of Vertex bases. The two on the left have adaptors for U.S. engines and the other is as-imported.

RIGHT—This is the rotating part of the portion of the Vertex that generates the primary current for the mag's induction coil.

FAR RIGHT—Breaker points for Scintilla Vertex magnetos have platinum contacts and are designed to function efficiently at high engine speeds.





FAR LEFT—The Vertex coil is shaped to fit inside the mag's housing where it is protected from moisture. Two leads are for primary current, the other for secondary.

LEFT—Wire sockets in Vertex caps are numbered in the order they receive current from the rotor. The arrow at the top of the cap indicates the direction of rotor rotation.

problem if the timing of the discharges at the spark plugs for maximum torque output didn't change with engine speed. If the timing requirement were the same at all engine speeds, a distributor or magneto could be adjusted to the correct timing and locked at that point. But because the timing requirement changes with engine speed, some provision must be made in the ignition system to automatically change the timing as the speed changes. This is done in both distributors and magnetos by governor weights that rotate with the breaker cam's shaft. Linkage between the weights and the cam rotate the cam in relation to the shaft when centrifugal force causes the weights to move. Rotating the cam in this manner causes it to open the breaker points at different times in relation to piston position in the cylinders.

One of the problems of obtaining maximum performance from an engine is determining the best ignition timing for different crankshaft speeds. At best this is a change and try proposition. The usual procedure is to adjust the weights in the distributor or magneto so they provide the timing advance curve previous experience has shown to be best for a particular engine and then try different initial timing settings until the engine performs its best. Things that determine the advance curve an engine should have are concerned with the design and dimensions of the engine's cylinders and combustion chambers, whether the engine is naturally aspirated or has a blower, the fuel used, etc.

It wouldn't be right to say that no two reworked engines are the same because there are many more engines than there are possible reworking combinations. However, there are enough possible combinations to make it necessary for the advance mechanisms of special distributors and magnetos to be adjustable if it is to be possible to make the advance curves of the ignition systems anywhere near correct. Methods of making the adjustments vary for the magnetos and distributors involved but they all have to do with the governor weight assemblies.

The advance curve that experience has shown to be best for a particular engine and the way it was reworked is built into a distributor or mag by its manufacturer. This means that, for best results, a distributor or mag should be ordered for a specific engine. This will give the ignition system its best chance of having the most satisfactory advance curve for the engine; however, this isn't any guarantee that the curve will be absolutely correct at all crankshaft speeds. This is something that can be determined only with expensive testing equipment and hours of experimentation on the actual engine.

A special distributor is only part of a battery ignition system. To do its job correctly the system must also have good coils and condensers, good primary and secondary wiring, and a good battery.

Coils are important because they must be capable of creating sufficient secondary voltage to fire the spark plugs. This isn't too difficult with a two-coil system but it is still important that good coils be used for maximum reliability. Condensers should be matched to the coils. Their duty is to reduce to a minimum the amount of arcing that occurs between the breaker points when the points are first opened. Too much arcing can slow coil action, lower the efficiency of breaker point contacts, and shorten contact life.

Primary and secondary wiring must be in good condition and have tight joints. Old, deteriorated, or poorly installed wiring can be the cause of excessive voltage drop in the primary circuit between a battery and the coil and in the secondary circuit between the coil and the spark plugs. As one of the main purposes of using a special battery ignition system is to create the required secondary voltage at the spark plugs, it would be extremely foolish to handicap the system with poor wiring.

The importance of a good battery to the system can be understood when it is realized that the battery is the source of the system's electrical output. It must be capable of delivering current of full voltage to the primary windings of the system's coils if the coils are to be able to create sufficient secondary voltage. This makes it imperative for the battery in a competition car fitted with a battery ignition system to be in a fully charged condition when the car is being driven in a competition event.

The generator on an engine used for street driving should keep the car's battery charged but for engines that don't have generators some sort of battery charger will have to be used. The best program to follow with a charger is to maintain the battery in a fully charged condition between its periods of use with a trickle charger that has an output of only a few amperes. A charger of this type, used over a period of several hours, will charge a battery much more completely than will a charger of the quick-charge type. Quick-chargers are all right for giving a battery a quick boost for the purpose of starting an engine and getting a car on the road again but they leave a battery only partially charged. Use a hydrometer to determine battery condition. A hydrometer reading of approximately 1.280 indicates a condition of full charge.

A magneto doesn't require primary wiring except for the wire from its primary terminal to a switch somewhere near the driver, but it must have secondary cables to connect it to the engine's spark plugs. Plug cables for magnetos can be the same as those for special battery ignition systems; however, under no circumstances should cables of the TVRS type be used on any competition engine. TVRS cables have a non-metallic resistance-type conductor designed to reduce television and radio interference caused by automobile ignition systems. The resistance in these cables makes them unsuitable for engines that may require at their spark plugs all the secondary voltage their ignition system can deliver.

The best secondary cables for competition engines have copper or stainless steel conductors. Any nationally advertised brand of cable that has a conductor of either metal should be satisfactory. Terminals on the ends of the cables necessary for their installation may be either soldered, crimped, or threaded but they must make positive contact with the cables' conductors.

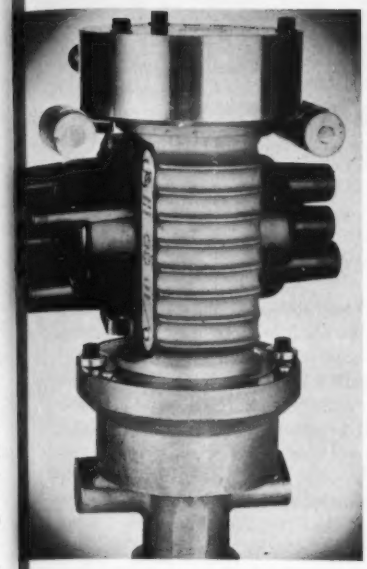
Spark plugs are a highly important part of any ignition system but their lives are short compared to those of the rest of the parts in the system. They must be changed frequently for best results. Things that determine the effectiveness of plugs are their heat range and their general condition. The heat range must be correct for conditions that prevail for a specific run. The plugs must be clean, both inside and out, and in nearly new condition. Some really hot cars can make only three or four runs on a set of plugs.

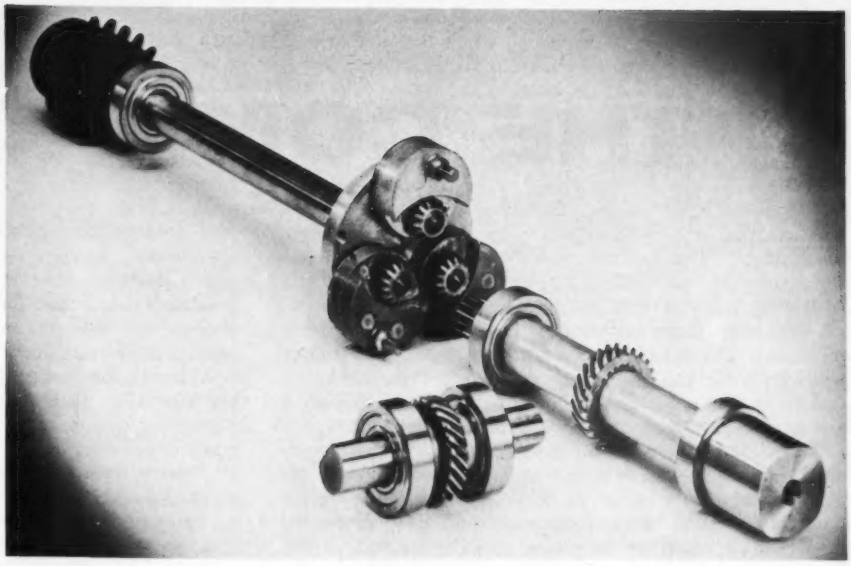
SCINTILLA VERTEX MAGNETO

In the hot rod world, the name Scintilla Vertex is almost synonymous with the word magneto. Vertex mags have gained so much popularity that they are practically standard equipment on highly modified engines used strictly for competition. Very rarely is a battery ignition system seen on an engine of this type.

Vertex mags are made in Switzerland. They are designed for automotive installations. Companies that distribute them in the United States make special bases for them to adapt them to the engines made in this country.

Simplicity of installation undoubtedly had much to do with the original acceptance of Vertex mags by hot rodders. All a fellow has to do to install one of them is remove his stock distributor and slip the mag into the distributor opening in the cylinder block. Ease of installation is still





ABOVE—These are the moving parts inside a Roto-Faze. The gear at the left is the drive gear. The breaker cam is on the end of the shaft at the right. Cross shaft rotates rotors.

in a Vertex's favor but now it is secondary to the flawless ignition and dependability it provides.

The advance mechanism in a Vertex is fully adjustable. The rate at which the advance curve climbs, as well as the curve's length, can be matched to the specification for any reworked engine. It is important that these adjustments be correct for the engine in which the mag is to be used.

Vertex mags have a high first cost, compared to that of battery ignition systems, but experience has shown them to have excellent resale values that battery ignition parts seldom enjoy.

BATTERY IGNITION DISTRIBUTORS

One of the two special battery ignition distributors worthy of mention is the Roto-Faze. This distributor was designed by Kong Jackson, a fellow who started building quality distributors years ago, but is now being manufactured by Warren Sanborn.

The Roto-Faze is a two-coil distributor constructed of the finest materials and workmanship. It differs from standard distributors by having two four-cylinder caps instead of a single eight-cylinder cap. Caps, rotors, and breaker points are standard automotive products. This makes it easy for a rodder to buy replacements at a local parts store when these parts become worn.

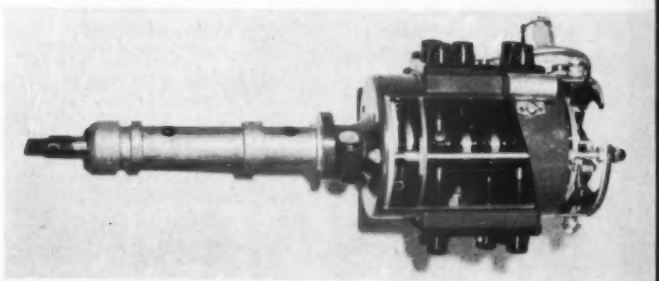
The advance mechanism in a Roto-Faze is adjusted at the time the distributor is assembled to provide the recommended advance curve for the engine on which the distributor will be used. Roto-Faze distributors do not have a vacuum advance diaphragm, such as is found on most standard automotive distributors. All the advance is effected by the governor weights.

Another excellent distributor is the "Flamethrower," which is manufactured by Tom Spalding, another fellow well-known for his ignition system products. Flamethrower distributors are also for two coils. The only standard parts they use are their breaker points and, on those that are so-equipped, their vacuum advance diaphragm. Their governor weight advance assembly can be adjusted to suit any engine.

Materials and workmanship in Flamethrowers are of the best. These guarantee accurate, positive ignition for either road or competition engines.

DON FRANCISCO

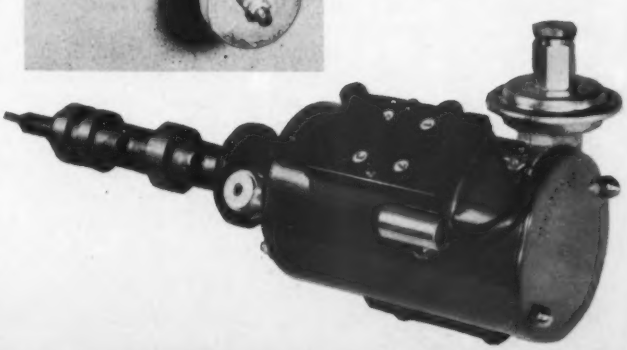
ABOVE, LEFT—The Kong "Roto-Faze" two-coil distributor was designed by "Kong" Jackson, a pioneer in the hot rod ignition system field. It is now manufactured by Warren Sanborn.



Display Flamethrower has sections cut from its housing and cap to enable its rotor and secondary pickup terminals to be seen.

LEFT—This is the latest of Mallory Electric Corp. coils. It is called the "Flash-Fire." It is available in 6- and 12-volt types.

BELOW—One of the very best of the special two-coil distributors is the Spalding "Flamethrower." This one has a tachometer drive.



THE COMPACTS

THE YEAR 1960 will go down in automotive history as "The year of the big gamble" for U.S. automobile manufacturers. The gamble involved literally millions of dollars' worth of time and machinery the manufacturers had invested in the small-size "compact" cars they hoped a certain percentage of new car purchasers would buy.

No industry as large as the group of companies that manufacture automobiles in this country takes a fling at any new project that costs as much as the new compact car program cost until exhaustive research has been conducted to determine whether the project will be successful. For the manufacturers interested in compacts the conclusions they gained from their research in favor of the compacts was supported by the sales figures of the so-called economy cars imported into this country from England, France, Germany, Italy, Sweden, etc. Enough of the imports were being sold to make small American cars practical from the sales standpoint if they could capture a great enough percentage of the small-car market.

Most persons who bought the small foreign cars based their decision on the high miles per gallon of gasoline figure claimed for them. They had visions of cutting their automobile operating costs to a third or even a quarter or what it was with their former full-size American automobile. But a great percentage of these purchasers soon found that the inflated costs of spare parts and labor to repair their cars, and the frequency with which repairs had to be made simply because the cars weren't capable of withstanding the rigors of normal U.S. driving without almost constant attention, made operating costs almost equal to that of a genuine automobile.

Another disadvantage of the little cars is that their performance is so poor that a person literally takes his life into his hands when he ventures onto a freeway or the open highway. They are so lacking in power that they can't accelerate well enough to merge with a line of fast moving traffic. On hills, where it is sometimes necessary to pass a slow moving truck, passing is entirely out of the question unless the road is wide and there isn't any other traffic on it. On mountain roads, where passing isn't allowed, long lines of cars slowly plodding up hills are a common sight. At the head of the line, chugging and grinding and grating away, usually will be found an imported economy car whose driver doesn't have the common sense or courtesy, or perhaps is so disgusted with his vehicle and himself that he has decided to let the rest of the world suffer with him, to pull over and let the automobiles go by.

The path for the U.S. manufacturers was clear. To be successful in the small-car field all they had to do was build small cars that were attractive and provided reasonably good fuel mileage, that performed well enough to make them safe and passably enjoyable to drive, and that didn't require repairing any more often than their full-size competition. That the manufacturers succeeded in meeting these requirements is proved by the steadily declining sales of imported small cars. Sales for some of the imports have dropped so low that many dealers and some importers have been forced out of business.

Although the first compacts were more appealing to new car purchasers than their manufacturers had hoped it was soon found that sales would be even better if the little cars could accelerate better. Their maximum speeds and cruising speeds were adequate but the time it took to get up to cruising left something to be desired, although it was only in the neighborhood of half that required by the imports. The obvious solution was to build a little more horsepower into the compacts. This was accomplished by making available high-performance kits for some of the '60 models and making optional engines with larger displacements and greater horsepower available in '61. The consumer who went for the better performance lost some fuel mileage but it was soon learned by those who were interested that fuel mileage was quite low on the list of features the purchaser of a small car wanted. Mileage, even with the high-performance options and the larger engines, was still so much better than that attained from a full-size car that it was a definite factor in favor of the compacts.

The impact compact engines have made on hot rodding hasn't been earth-shaking but the potential they have as power plants for all sorts of small-displacement competition classes is intriguing. As of this time the most notable successes have been the Falcon prepared by Bill Stroppe and his crew for Bill Burke's 200 plus mph streamliner and the four-cylinder Tempest in one of Mickey Thompson's dragsters. As more ready-made equipment becomes available for them, more of the engines will be seen in competition cars. The purpose of this chapter is to present a compilation of what rodders have done so far to various compact engines and of the special equipment available for them.

FALCON-COMET

Ford Motor Co. uses the same six-cylinder in-line engine in their Falcon and Comet compacts. In 1960 it had 144 cubic inches of displacement and was rated at 85 horsepower. Its bore was 3.50 inches and its stroke was 2.50 inches. In 1961 the '60 engine was retained as standard equipment and a 170-inch version was a factory-installed option. Horsepower of the 144-inch engine remained at 85; the 170 inch version was rated at 101 hp. The larger engine used the same cylinder block and bore diameter as the smaller one but its stroke was lengthened to 2.94 inches, which made it .440 inches longer than that of the standard engine. The compression ratio of all engines was rated at 8.7 to 1.

Falcon-Comet engines have proved to be as sturdy as any of the full-size automobile engines. They will push a passenger car all day long and day after day as fast as any intelligent man cares to cruise on modern highways. They do not have any inherent faults that might make them

unsuitable for competition conversions. The only thing not in their favor is the design of their induction system. This makes the installation of additional carburetor venturi area difficult, but not impossible.

To reduce production and assembly costs the intake manifold for these engines was designed to be an integral part of their cylinder head. The manifold consists of a log of tubular shape attached to the left side of the cylinder head with a short branch for each intake port. Midway between the log's ends is a riser for a single-throat carburetor. Because the engine was designed primarily to deliver good fuel mileage rather than its maximum potential torque and horsepower outputs, the log was given a small cross-sectional area. A small manifold improves economy by keeping mixture velocities high. High velocity improves mixture distribution between the cylinders and keeps the fuel in suspension in the air but a small manifold limits the quantity of mixture that can flow from the carburetor to the cylinders. This limits the engine's torque and horsepower outputs. To compensate somewhat for the larger displacement of the 170-inch optional engine the intake log on its cylinder head has a slightly larger inside diameter than the 144-inch log.

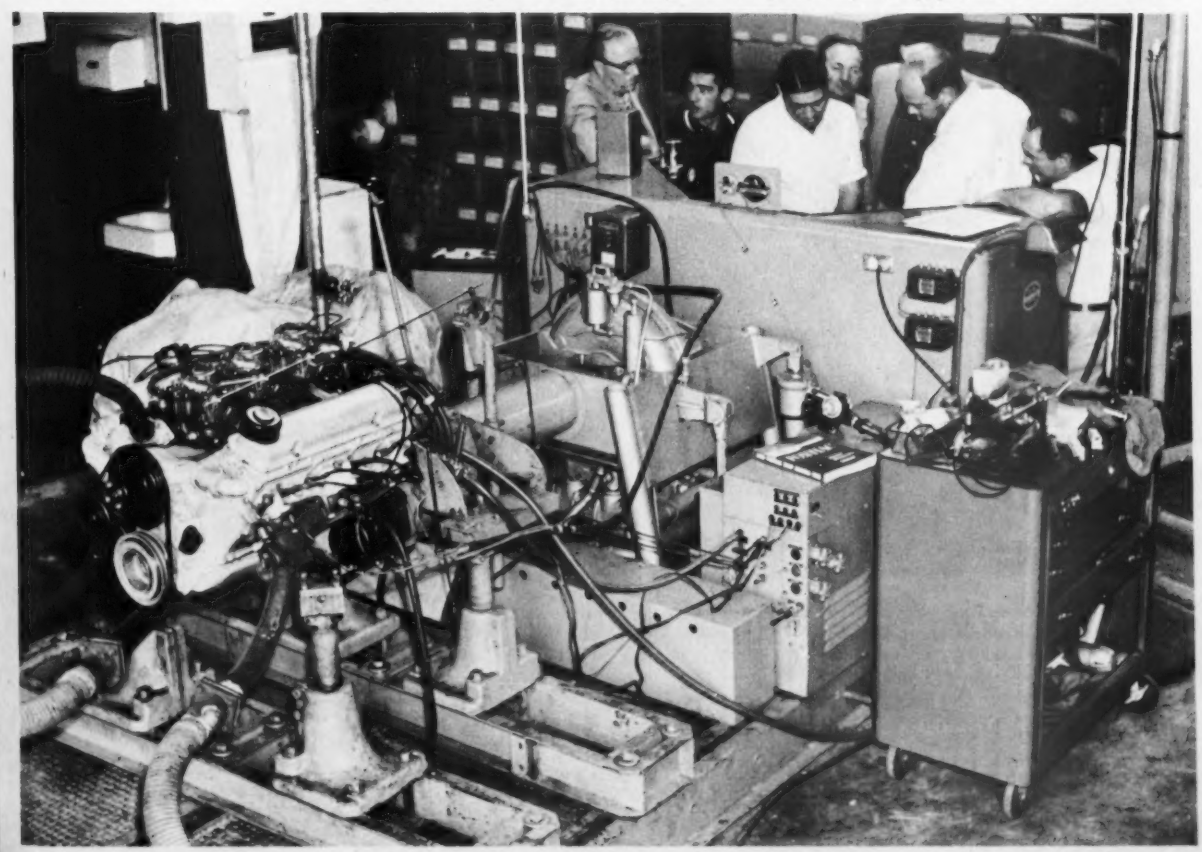
It is absolutely essential that the venturi area of a Falcon-Comet engine's carburetion system be increased if more power is to be obtained from the engine. Several different ways of doing this have been devised. One, which is the simplest but the least effective, is to enlarge the inside diameter of the riser on the standard manifold log to match the throat in a single-throat carburetor that has a larger venturi than that in the standard carburetor. A carburetor used for this sort of modification is the one for the standard-size Ford 6. Its venturi diameter is ⅛-inch larger than that of the standard Falcon-Comet carburetor. A variation of this modification is to enlarge the manifold riser and install an adaptor flange and a two-throat carburetor. A Holley 94 or a Stromberg 97 are the usual choices for these installations. Performance with one of them will be greatly improved but the mileage for normal driving will not suffer greatly.

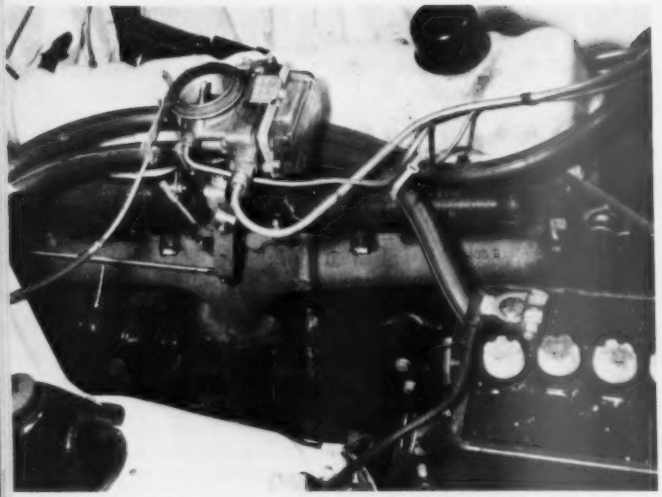
A carburetor that has greater venturi area on the standard manifold log can help performance but it isn't the best approach to the problem because flow to the cylinders is still restricted by the log's small inner diameter. The effect of the second method of increasing venturi area is also limited to a certain extent by the log's inner diameter but not nearly as much as with the first method. It involves the installation of a special adaptor and three Falcon-Comet carburetors. Adaptors for this purpose that achieve the same result in the same manner were designed and manufactured by both Bill Stroppe and his crew and Edelbrock Equipment Co. at almost the same time. Now, the adaptors are available from other manufacturers.

The three-carburetor adaptor consists of an aluminum member that has three carburetor mounting flanges on its upper surface and a single flange midway between its ends on its lower surface. When it is placed on top of the manifold log so its middle flange is in line with the log's flange, the ports in its ends are directly over the log at points between cylinders one and two and four and five. Its underside, around the end carburetor ports, is curved to match the curvature of the log.

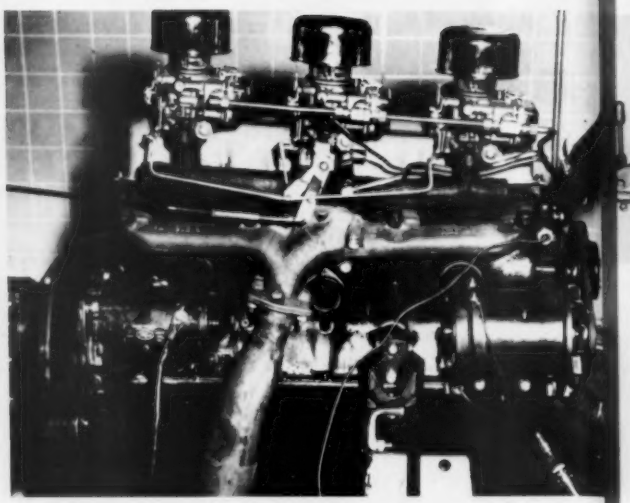
The adaptor is installed by first removing the cylinder head from the engine and then cutting a port in line with

Vic Edelbrock uses this dyno in his shop in Los Angeles for the development of special engine equipment he manufactures.

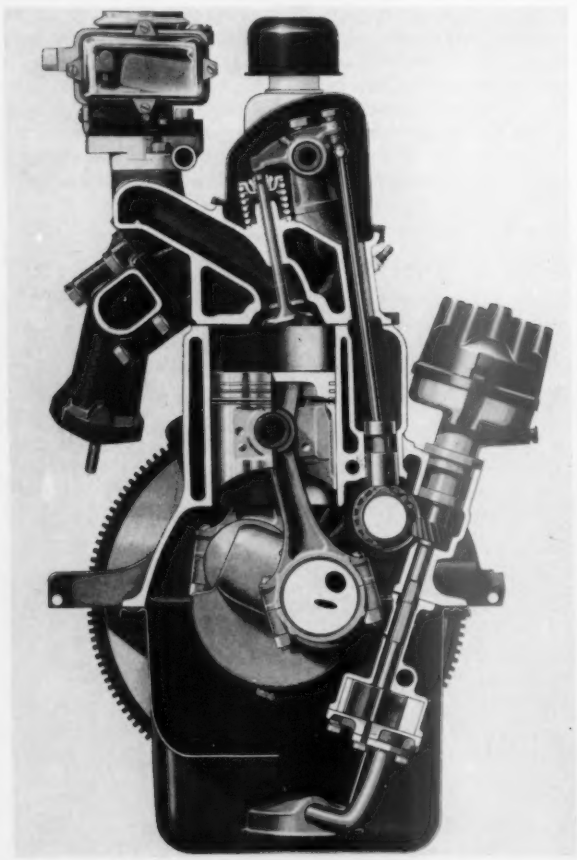




Ford's Falcon-Comet is a little six-in-a-row chugger that has plenty of stamina and performance for everyday usage.



Carburetor venturi area on this Falcon has been tripled by the installation of two more standard carburetors on its manifold.

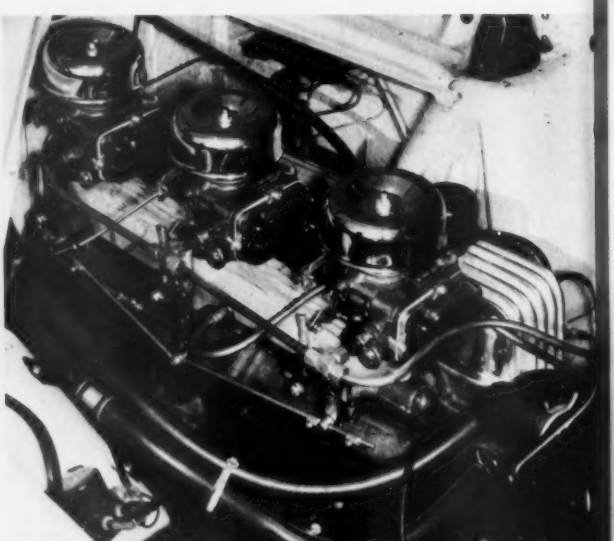


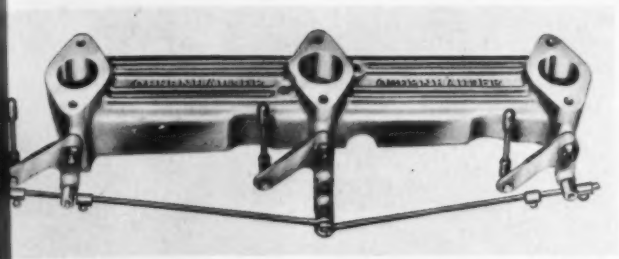
Cross-section of the Falcon-Comet engine shows its simple and sturdy construction. There isn't anything unconventional about the engine to make it difficult to repair.

This Falcon has also been fitted with three carburetors but in a different way. Instead of welding risers to the manifold, a special aluminum adaptor was used.

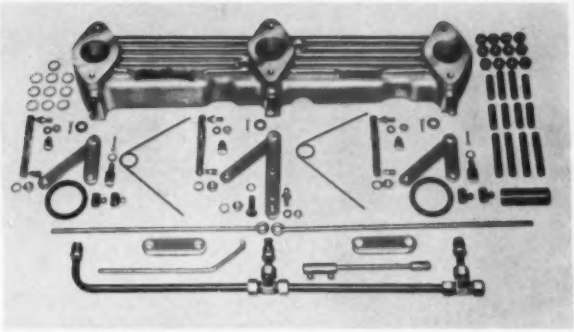
each of the adaptor's end ports through the top of the log with a hole saw. Areas around the ports are then filed smooth to provide suitable seats for O rings in the adaptor that seal the joints between it and the log. Special clamps that secure the adaptor's ends to the log and the end carburetors to the adaptor are then installed. The middle of the log is secured to the manifold with extra-long studs that also secure the middle carburetor to the adaptor. Installing three carburetors in this manner triples the engine's carburetor venturi area while at the same time reducing the restrictive effect the log's small internal diameter has on the flow of mixture to the cylinders by shortening the distance from the carburetors to the cylinders.

Fuel mileage equivalent to that with a stock single carburetor for all normal driving is possible by using progressive throttle linkage on the carburetors. With this linkage only the middle carburetor is used for normal

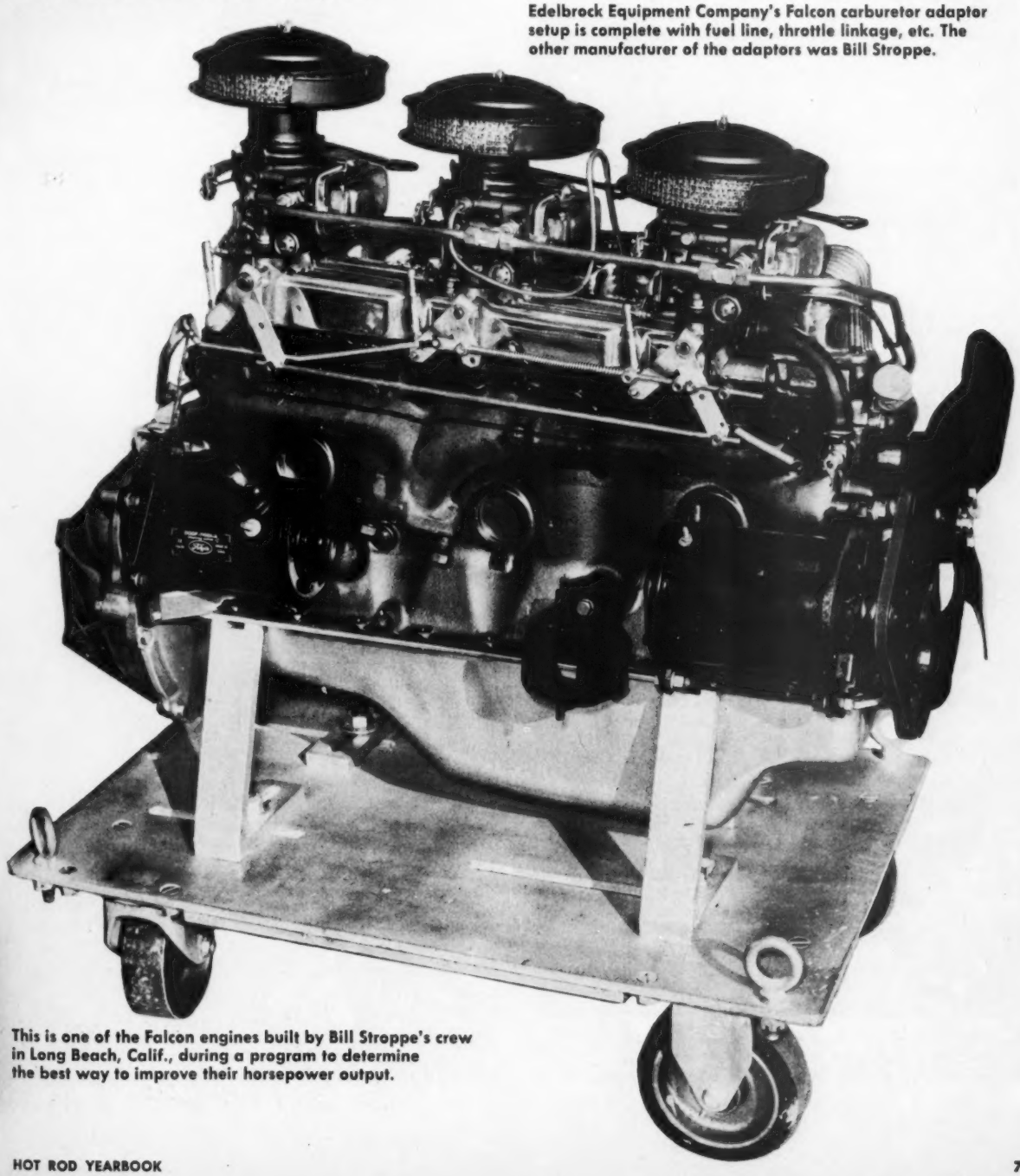




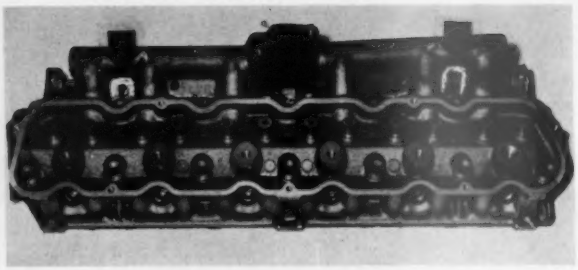
At least three different companies designed and manufactured carburetor adaptor kits for Falcon-Comet engines. This one is by Offenhauser Equipment Corp., L.A.



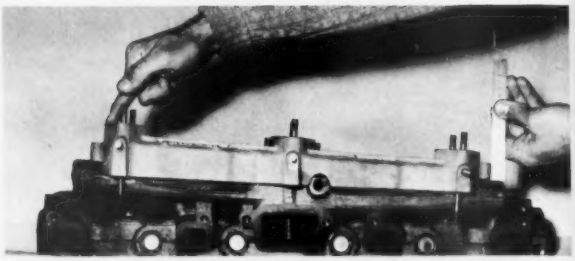
Edelbrock Equipment Company's Falcon carburetor adaptor setup is complete with fuel line, throttle linkage, etc. The other manufacturer of the adaptors was Bill Stroppe.



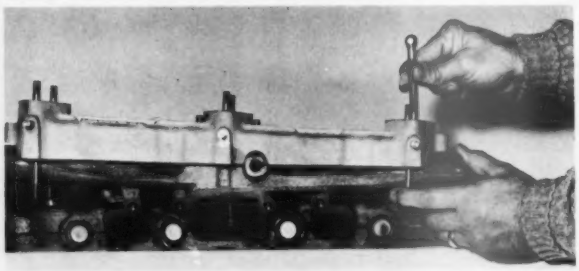
This is one of the Falcon engines built by Bill Stroppe's crew in Long Beach, Calif., during a program to determine the best way to improve their horsepower output.



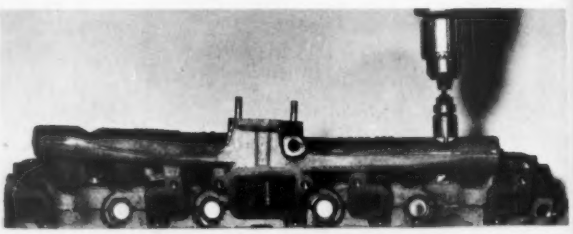
First step in the installation of a three-carburetor adaptor on a Falcon-Comet is to enlarge the marked holes for the studs or bolts that will secure the adaptor.



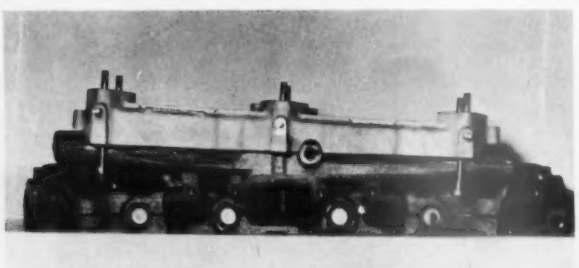
The second step is to fit the adaptor to the manifold so the O rings in its ends can make an airtight seal.



With the adaptor in the position it will occupy, the outlines of its end ports are scribed in the top of the manifold.

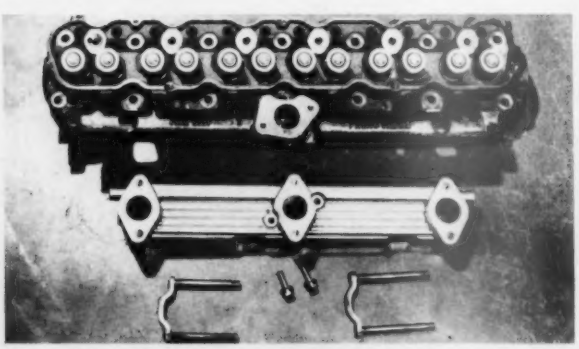


Ports in the top of the manifold can be cut with any available means but the easiest way is with a hole saw.



After the ports have been cut, the adaptor is realigned on the manifold and clamped in place with steel straps and nuts.

One of the O rings shown in this photo is inserted in the groove around each of the adaptor's end ports. The surface of the manifold is filed smooth to make a seat for them.



This is the finished head and manifold assembly and the adaptor. The U-bolts with this adaptor differ from the studs in the other photos. These also hold the carburetors.

cruising speeds. The throttle valves in the end carburetors start to open when the middle carburetor's throttle lever reaches a pre-determined position. This makes all three carburetors readily available for high speeds or the acceleration required for passing.

One of the problems created by installing three carburetors or a special carburetor of some type on one of these engines is that the standard Falcon-Comet ignition distributor will function correctly only with one standard carburetor. The distributor is the old Holley design that was used for years on full-size Ford engines. It utilizes vacuum that is a combination of a low pressure created by air flow velocity through the carburetor's venturi and the low pressure in the engine's intake manifold to advance the ignition timing. Increasing the carburetor venturi area in any manner changes the pressure that acts on the distributor's advance diaphragm. This makes it impossible for the distributor to advance the ignition timing at the correct rate. The result is that the timing is always slower than it should be.

The only practical solution to the ignition system problem is to replace the distributor with one that has a mechanical advance mechanism. Distributors of this type are available from Autolite, which now is the Motorcraft Division of the Ford Motor Co., and Mallory Electric Co. An engine to be used strictly for competition would benefit from the installation of a Scintilla Vertex magneto. These are also available for these engines.

The third carburetion improvement is strictly for competition. It is the installation of a Hilborn fuel injector. The installation requires considerable machine work on the cylinder head but once the job has been done the results are terrific.

Stroppe installed a Hilborn injector on the engine Burke used in his streamliner. The basic problem was to remove the standard manifolding and provide a suitable surface to which the injector could be bolted. The first step was to cut the log off the head and machine the ends of the intake ports flush with the exhaust manifold flanges. A plate that would cover both the exhaust ports and the intake ports was then cut from 5/16-inch steel plate. Openings to match the ports were cut in the plate and the plate was furnace brazed to the head. After the head had cooled, all its machined surfaces were remachined to make them flat again by eliminating any distortion that might have resulted from the brazing operation. The head now had a flat surface to which an injector and special exhaust pipes could be bolted. Suitable holes were drilled in the plate and threaded for the capscrews that would secure the parts to the head.

After installation of the plate had been completed, the head's intake and exhaust ports and passages were enlarged as much as possible to simplify the engine's breathing problems. All valve seats were enlarged .090-inch and lightened and polished oversize valves that matched the seats were installed. Surfaces of the combustion chambers adjacent to the valve seats were ground to enlarge the chambers for the larger valve heads and the surfaces were polished.

From all indications cylinder blocks for all '60 and '61 engines are identical. Cylinders in both blocks have been bored 1/8-inch oversize and used without trouble. Whether or not this is the practical maximum for boring won't be known untill a few cylinders are destroyed by boring bars or a few bored cylinders crack or break while in use.

Crankshaft stroking is simplified for a 144-inch engine by installing a shaft from a 170-inch engine in it. The 170 shaft can be installed without any alterations to either it or the cylinder block. Whenever a 170 crankshaft is used, 170 connecting rods should also be used. These are identical to 144 rods except that their center to center length of 4.715 inches is .140-inch shorter than that of the 4.855 inches of 144 rods. Using shorter rods will reduce the amount the compression height of the pistons will have to be shortened for the longer crankshaft stroke.

If pistons made for a 170 were used with the 170 crankshaft and connecting rods but with a 144 cylinder head, the combustion chambers in the heads would give the cylinders a compression ratio of approximately 9.7 to 1. However, a 170 head is more desirable because of its larger intake valves and intake manifolding. The combustion chambers in 170 heads have a volume of 53 cubic centimeters so that in combination with the 170's larger cylinder displacement they will provide the same compression ratio of 8.7 to 1 that the 144's have. This, then, is the easiest method of modernizing a 144 that has cylinders for which standard or factory replacement pistons are available. The compression ratio can be boosted by milling the cylinder head. Milling any of the heads .060-inch boosts the compression ratio of cylinders that have their standard displacement to approximately 9.4 to 1.

For cylinders larger than those for which factory replacement pistons aren't available it will be necessary to obtain special pistons. These are available for all practical bore diameters and stroke lengths. The compression height of pistons of this type can be juggled by their manufacturer to provide any desired compression ratio within practical limits. For the higher competition ratios, the pistons are fitted with domes that protrude into the cylinder head's combustion chambers when the pistons are at top center in their cylinders.

While the engine is apart it's a good idea to have the balance of its crankshaft and rod and piston assembly checked. For an in-line engine such as a Falcon-Comet the weights of the pistons and connecting rods don't have any effect on the crankshaft's balance as they do in a V8, but all the pistons should weigh the same, all big-ends of the connecting rods should weigh the same, all small ends of the rods should weigh the same, and the rods' total weight should be the same. The important thing about the crankshaft is that it be in balance along its entire length so that no part of it will try to move away from its axis when it is rotating at high speed. The engine's flywheel should be balanced while it is mounted on the crankshaft or on a special mandrel.

Heavy-duty bearing inserts should be used for the crankshaft's main bearings and in its connecting rods. Clevite 77 inserts, which have copper-lead alloy bearing surfaces, are available. The bearings should be fitted with the maximum clearances allowed by factory tolerances. This will guarantee minimum friction in the bearing assemblies and ample oil flow to not only lubricate and cool them but also to adequately lubricate the rest of the engine's internal parts.

Reground camshafts and installation kits are available for Falcon-Comets from most camshaft regrinders. One of these is essential if any appreciable horsepower increase is to be realized from the engine. Included in the installation kit are special valve lifters necessary if maximum wear is to be realized from the camshaft, and valve springs that are stiffer than stock springs to make the valve action follow the cams on the camshaft at high crankshaft speeds.

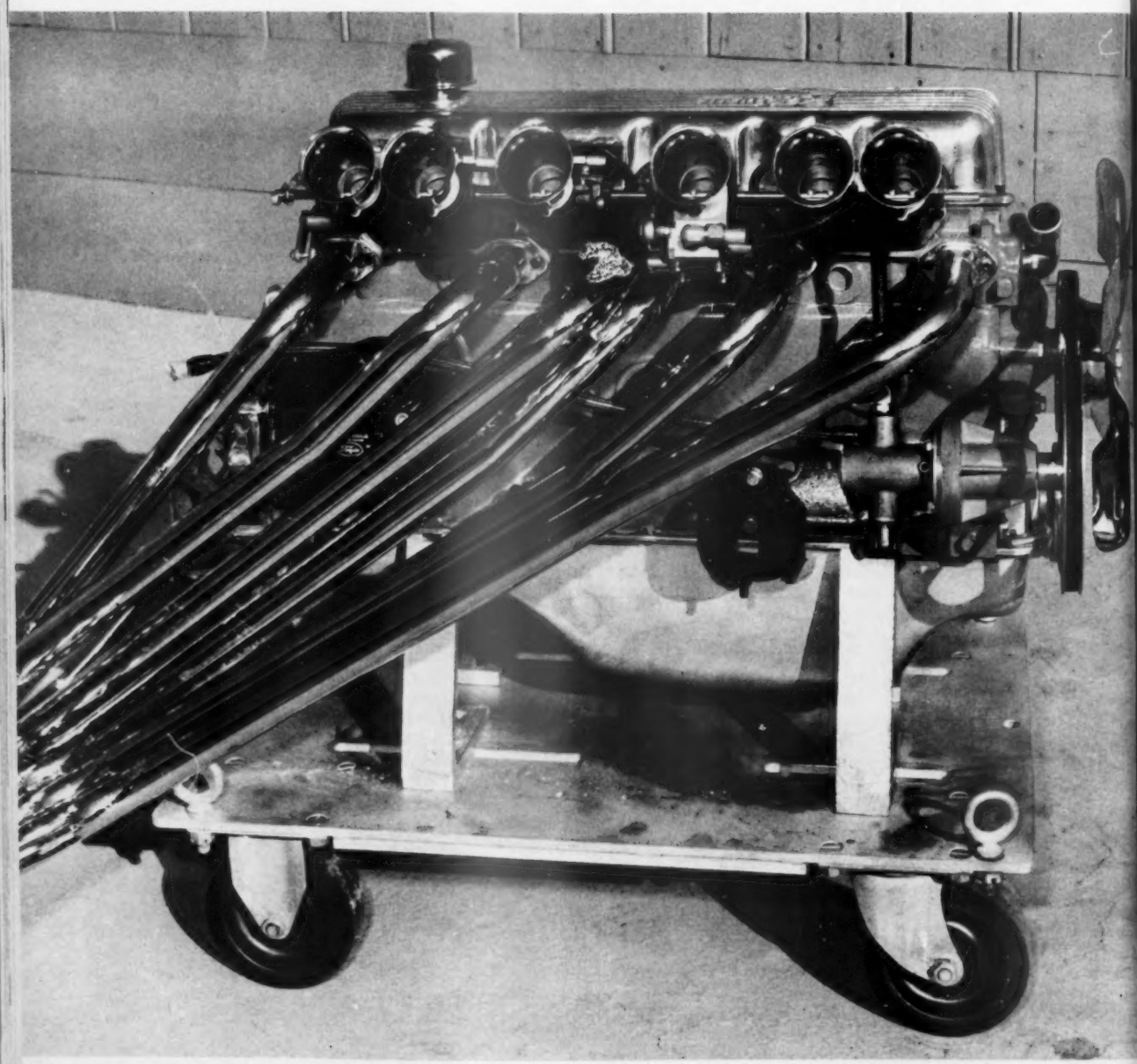
When selecting a reground camshaft take the advice of the man from whom you are buying it because he will know how the camshaft will make the engine perform.

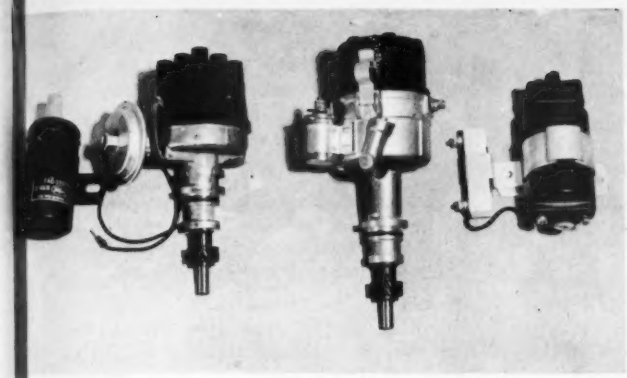
GENERAL MOTORS' ALUMINUM V8'S

Fellows who were reworking flathead Ford V8 and Mercury engines in the late 1940's often remarked how much they would like to have an engine similar to the flatheads but with five main bearings and overhead valves. These fellows got their wish in 1955 when Chevrolet began selling their wonderful little V8. Now, six years later, someone has rubbed the magic lantern again and the General Motors Genie has once more made magic motions with his hands and produced two more engines that would have fulfilled the wishes of the flathead group. But this time the Genie has gone a step farther by making the engines' cylinder blocks and heads of cast-aluminum alloy rather than cast iron; however, he didn't hold his fingers quite right on one of his magic motions because his new engines are a little shorter on cubic inch displacement than the old flatheads. However, Genie may have been smarter than a first impression would indicate by producing engines of a design and displacement that may start an entirely new trend among competition hot rodders.

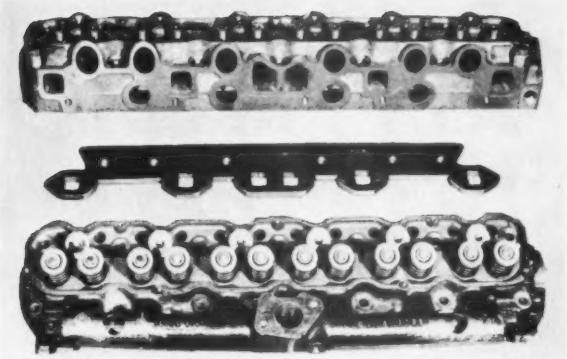
Genie's new engines are the V8's that are standard equipment in the Buick Special and Olds F85 and optional in Pontiac's Tempest. Although they are sold under different names, there are only two engines. One is used in the Buick and Pontiac and the other is for the Olds. Basically they are identical but they have different cylinder head

This Falcon, also built by Stroppe's crew, has a Hilborn fuel injector in place of standard integral manifold and single carburetor.

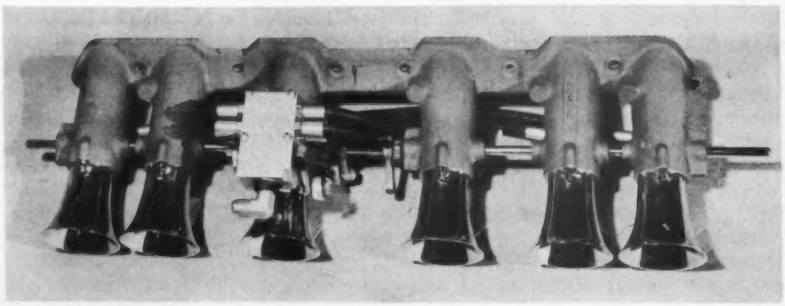




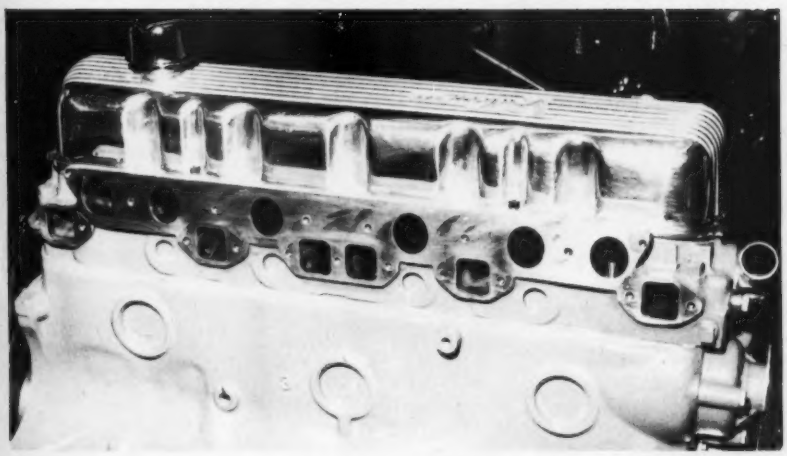
ABOVE—The Falcon distributor can't be used with three carburetors. The Mallory at the right makes a good substitution.

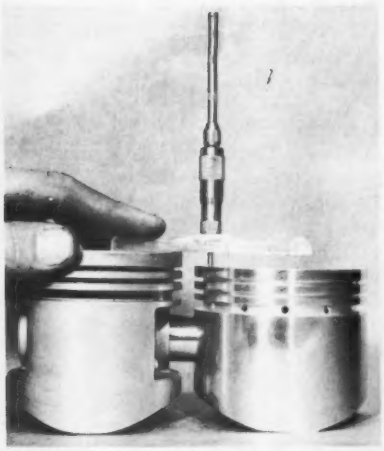


ABOVE, RIGHT—The standard manifold has been machined from the Falcon head at top. Steel plate will be brazed to it.



RIGHT—Stu Hilborn's Fuel Injection Engineering Co. made this injector for Stroppe but now you can buy one.



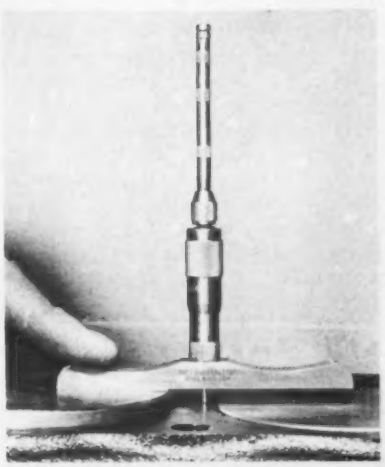


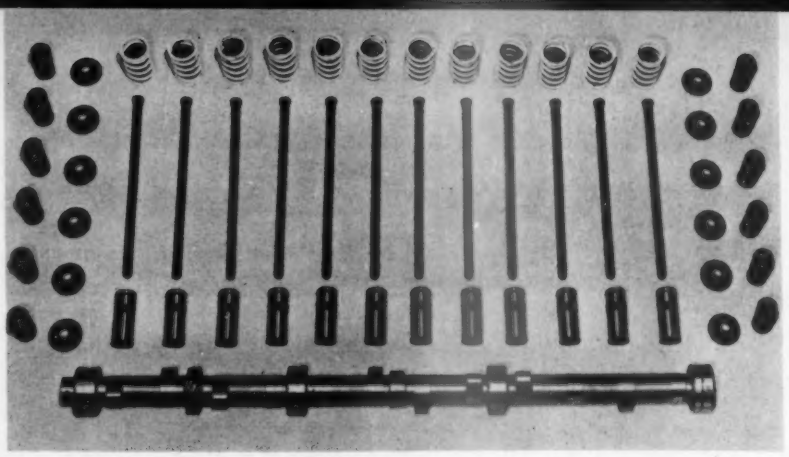
ABOVE, LEFT—Preparing head for the injector was quite a job, as you can see for yourself, but the result was worth the effort.

LEFT—Piston and rod bolt and nut at the left are standard parts from a Falcon. Those at the right are special for Stroppe's engine.

ABOVE, RIGHT—Compression height of special JE piston, right, is lower than that of stock piston to compensate for stroked crank.

RIGHT—Longer stroke of crankshaft from 170 Falcon lifts 144 piston on 144 rod .205" above top of either 170, 144 block.





All of the more popular cam grinders have grinds for Falcon-Comet engines. This one is Ed Iskenderian's Hard-face track grind, said good for 8000 rpm.

assemblies and pistons. Their cylinder blocks have identical internal dimensions and vary only in minor detail in external features at their bellhousing ends.

The most interesting thing about the engines is their aluminum cylinder block and heads. The block has a deep crankcase that extends below the crankshaft bearing bore centerline to give it greater rigidity. It has cast-iron cylinder liners that have ridges around their outer circumference to lock them firmly in the block. The liners are installed by placing them in the mold in which the block is cast and pouring the molten aluminum around them. Main bearing caps are steel. Cylinder heads have valve seat inserts of either steel or iron, depending on whether the engine is of Buick or Olds origin, for both intake and exhaust valves and cast-iron valve guides.

The combination of 3.50-inch diameter cylinders and a crankshaft stroke of 2.90 inches provided by the cast nodular iron crankshaft give the engines a displacement of 215 cubic inches. Factory horsepower ratings are 155 at 4800 rpm for the Olds and 155 at 4600 rpm for the Buick. Maximum torque output of the Olds is said to be 210 pounds-feet at 3200 rpm and 220 pounds-feet at 3200 rpm for the Buick. Compression ratios are 8.75 to 1 for the Olds and 8.8 to 1 for the Buick.

From the piston pins down, the engines are identical. Their connecting rods are fitted with Moraine 100-A bearing inserts for the 2-inch diameter crankpins on the well-counterbalanced five-main bearing crankshaft. Main bearing journals on the shaft are 2.3 inches in diameter. The crankshaft drives a sprocket and chain assembly that rotates a camshaft designed for hydraulic valve lifters and mounted in the conventional position above the crankshaft. A gear keyed to the camshaft's nose and secured to the shaft with the same capscrew that secures the drive sprocket drives the engine's distributor and oil pump.

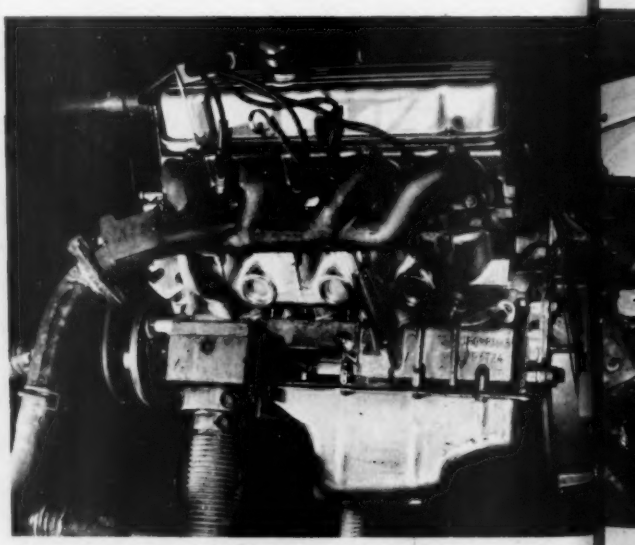
Pistons for the two engines are interchangeable as far as their mounting on the connecting rods and fitting the cylinders are concerned but their heads are shaped differently to provide the correct combustion chamber volumes when used in combination with their respective cylinder heads. Heads on the Olds pistons are flat but the Buick pistons have a concave, or recessed, head.

Where the engines differ is in their cylinder heads. Both heads are cast-aluminum but those for the Olds have the characteristic Olds wedge-type combustion chamber and those for the Buick conform with Buick practice by having a modified hemispherical chamber. Valves in both heads are in line but they vary a little in their head diameters. The Olds' valves are larger, the intakes having heads 1.522 inches in diameter compared to the 1.500 for the Buick intakes, and the exhausts are 1.353 inches in diameter compared to the 1.313 inches for the Buick's. The valves also differ enough in their stem diameters and stem lengths to make it impractical to try to interchange them. Valve springs differ in configuration, those for the Olds having a smaller diameter at their upper end and those for the Buick being of convential straight design.

Rocker arms for both engines have a ratio of 1.6 to 1 but for the Olds they are steel and for the Buick they are forged aluminum. Valve ends of the aluminum arms have hardened steel buttons that contact the ends of the valve stems. Stands for the Olds rocker arm shafts are secured with cylinder head capscrews, as they are in the big Olds engines. Buick stands are secured with shorter and smaller capscrews.

Another difference in the cylinder heads is that those on the Buick use only 14 of the 18 cylinder head capscrew holes in the cylinder banks but those for the Olds use all 18 holes. This gives the Buick heads five capscrews around each cylinder and the Olds heads six. Head gaskets for both engines are the embossed steel type that are slowly replacing other types.

Although the cylinder heads have different combustion



Olds F-85 on the dyno at Edelbrock's. Stock exhaust manifolds were reversed to simplify routing exhaust out of the building.

chamber and passage sizes, their port layouts and sizes are identical. This means that their intake and exhaust manifolds are interchangeable as far as bolting to the heads are concerned. The intake manifolds differ because of air cleaner requirements and the exhaust manifolds have different outlet flanges for their headpipes.

Each of the engines, complete with flywheel and all accessories, weighs approximately 325 pounds. This is pretty light for a stocker but the weight can be made even less by the installation of special high-performance parts.

From the standpoint of special parts availability these engines show great promise. Already available for them are reground camshafts in both flat-tappet and roller-tappet types, high-compression pistons, multiple-carburetor intake manifolds, exhaust headers, two-coil ignition distributors, and aluminum flywheels. Reliable information isn't available as yet regarding the engines' boring and stroking potentials but these are being explored now by some engine builders.

Edelbrock Equipment Co. conducted a series of tests with both engines on their engine dynamometer early in '61 to become familiarized with them and to learn something of their potentials. The results of these tests should prove interesting to the owner of a passenger car that has one of the engines and to fellows who are contemplating one of them for some sort of competition. Condensed results of the Edelbrock tests are listed in the accompanying table.

An inspection of the engines' innards after the tests showed that their main and connecting rod bearings were in excellent condition. The only engine part that caused any trouble at all during the tests was the distributor-oil pump drive gear in the Olds. It was learned from the mechanics in a local Oldsmobile agency that this difficulty was being experienced with many of the engines.

One thing Edelbrock's men learned the expensive way during the tests is that intake passage walls in the Olds cylinder heads aren't very thick. While trying to enlarge the passages a hole was quickly ground through one of their walls. A careful inspection of the head after this happened led the fellows to believe that the maximum thickness of the layer of material that can be safely removed from the walls is 1/16-inch. It is assumed that this limitation also applies to Olds exhaust passages and the intake and exhaust passages in Buick heads. After all the trouble the porting job caused it was found that it didn't help the engine's horsepower enough to be worth the effort and risk involved. So, until more is learned about this condition, it would probably be a good idea to forget about trying to port these heads.

The results obtained in Edelbrock's tests indicate an active future for the little V8's in hot rod circles. They may prove to be ideal for certain types of dry lakes, Bonneville, and sporty-type cars. They also hold considerable promise as good highway performers in their standard chassis when correctly reworked.

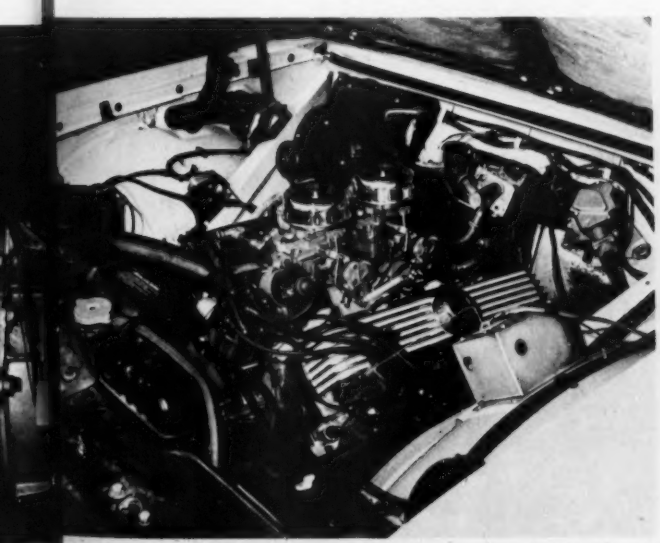
TEMPEST 4

Pontiac's Tempest four-banger compact engine is a genuine chip off the old block. It is actually one of their big V8's from which the left cylinder bank was chopped by a bit of surgery in the pattern department of the foundry that casts Pontiac cylinder blocks. Removing four of the cylinders made a new crankshaft and intake manifold necessary but the rest of the parts are interchangeable with those in the V8's.

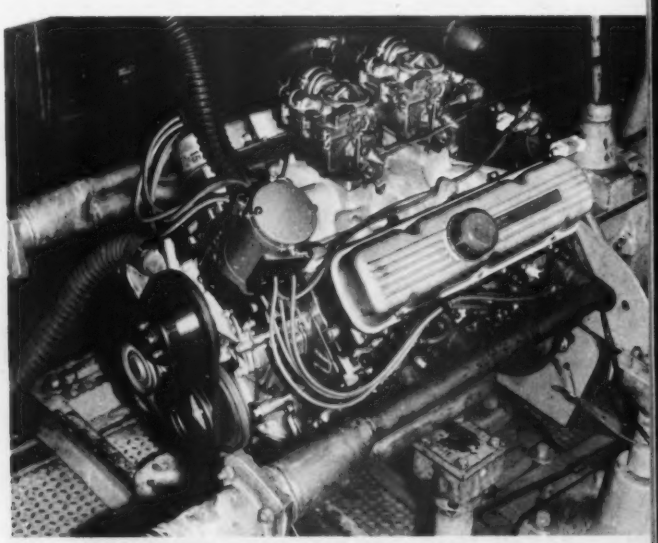
Bore and stroke dimensions in the four-banger are 4.06 and 3.75 inches, just as they are in the V8's, and displacement is 194.5 cubic inches, exactly half of the V8's displacement. Valves, connecting rods, piston pins, rocker arms, pushrods, etc., are the same as those in the V8's.

The list of factory-installed optional four-banger engines is almost as long as it is for the V8's. The basic engine has a compression ratio of 8.6 to 1 and a single-throat carburetor. Its horsepower is rated at 110 at 3800 rpm. The next engine on the list is identical to the basic engine except that its compression ratio is 10.25 to 1. Its horsepower is rated at 120 at 3800 rpm. Its torque is 200 pounds-feet at 2000 rpm.

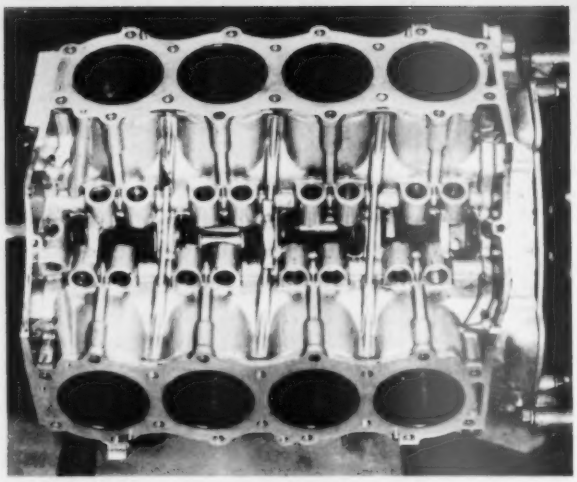
The second option has the basic engine's 8.6 to 1 com-



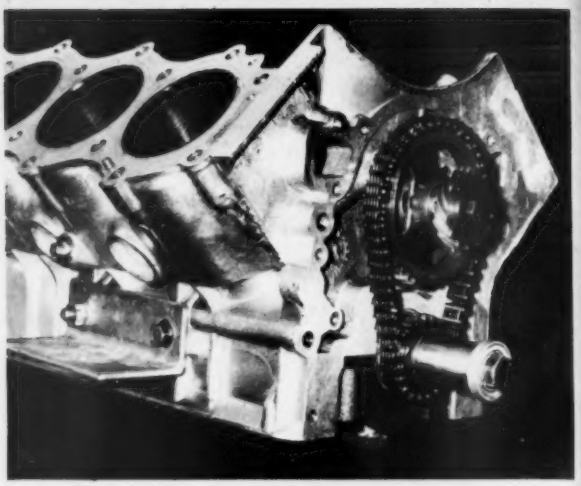
Olds F-85 that was used for test purposes in Vic Edelbrock's car. At this stage it had dual carburetors, aluminum covers.



F-85 with Edelbrock two-carburetor manifold, Chevy Rochester carburetors, Hedman headers, Spalding Flamethrower ignition.



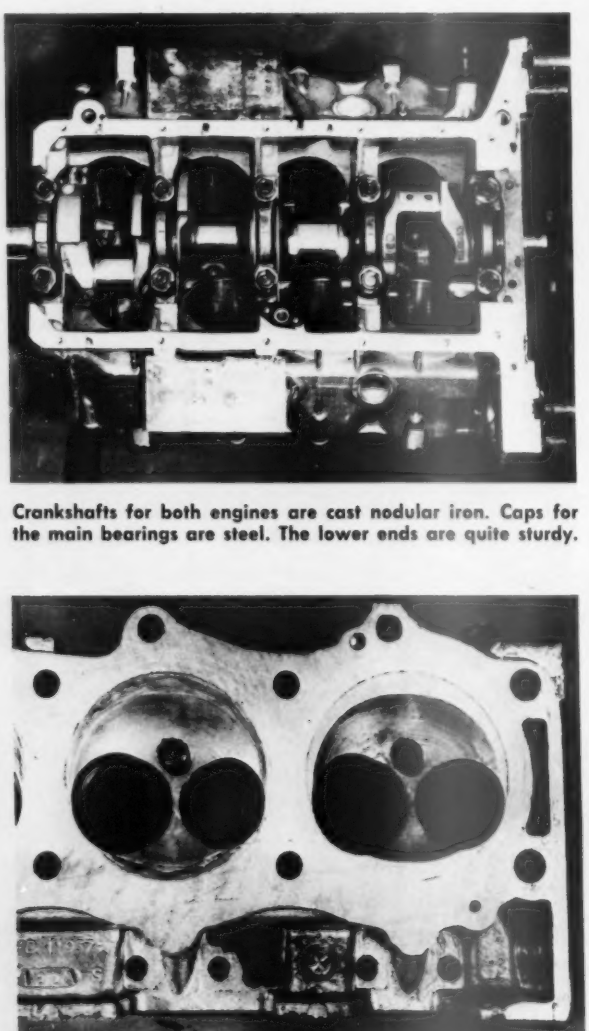
Cylinder blocks for the Olds F-85 and the Buick Special are almost identical. They have cast-in cast-iron cylinder liners.



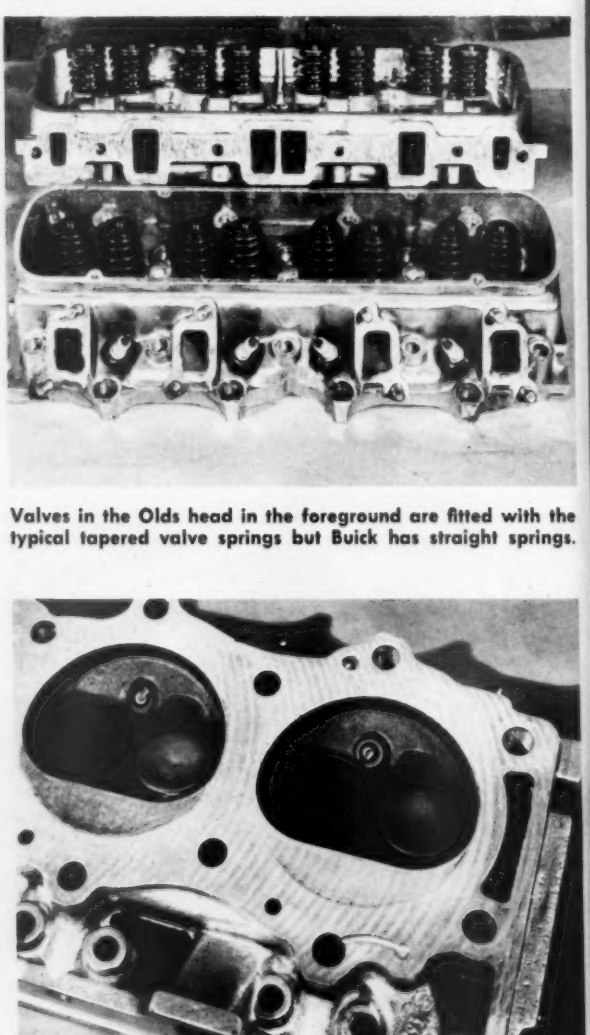
The crankcase on the aluminum cylinder block extends well below the crankshaft bearing bores to make the block stronger.

Crankshafts for both engines are cast nodular iron. Caps for the main bearings are steel. The lower ends are quite sturdy.

Valves in the Olds head in the foreground are fitted with the typical tapered valve springs but Buick has straight springs.



Cylinder heads for the engines vary radically in the design of combustion chambers. Buick stuck with semi-hemispherical.



Olds retained the wedge-type chamber they use in their larger engines. Valve head diameters of the engines don't vary much.

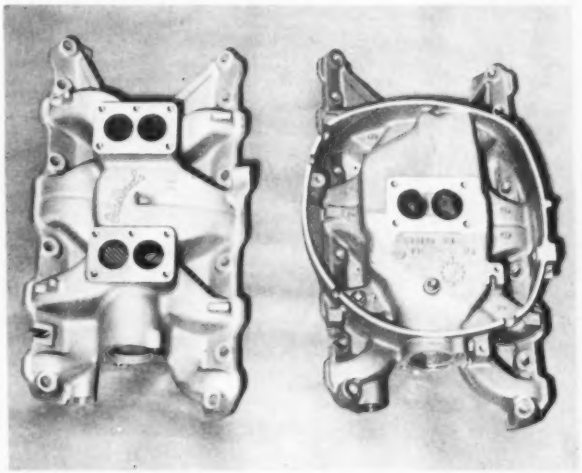
OLDS F85

Test #1 Completely stock, with 6½ hours of dyno break in time. No fan or generator but the water pump connected. Standard .046-inch carburetor main metering jets replaced with .048-inch jets. Thirteen degrees initial timing lead, 22 automatic degrees in the distributor. Actual compression ratio, 8.60 to 1.	135 hp at 4500 rpm
Test #2 Installation of a Chevy 283 two-throat carburetor that has 1 3/32-inch venturis in place of the Olds Rochester that has two 1-inch venturis.	140 hp at 4500 rpm
Test #3 Edelbrock two-carburetor intake manifold and two stock carburetors with .049-inch main metering jets.	156 hp at 4500 rpm
Test #4 Special JE pistons that raised the compression ratio to 12.30 to 1. Otherwise, the engine was exactly as it was for Test #1.	150 hp at both 4500 and 5000 rpm
Test #5 Special JE pistons and Edelbrock two-carburetor intake manifold and stock carburetors. Champion J 63R spark plugs, which are colder than standard plugs.	172 hp at 5000 rpm
Test #6 Isky E-4 flat-tappet camshaft and tappets, adjustable pushrods, and dual valve springs. Spalding two-coil distributor. Ignition reset to provide 32 total degrees of advance at 4000 rpm.	194 hp at 5500 rpm
Test #7 Iskenderian E-2 camshaft and stronger valve springs.	192 hp at 5500 rpm
Test #8 Hedman exhaust headers. Otherwise, engine exactly the same as it was for Test #7.	201 hp at 5500 rpm and 200 hp at 6000 rpm
Test #9 After Test #8 it was found that wear on the engine's distributor drive gear had retarded the ignition timing 3 degrees. The timing was corrected to a maximum of 32 degrees and the test was rerun.	213 hp at 6000 rpm
Test #10 Standard Old carburetors replaced with Chevy 283 two-throats that have 1 1/32-inch venturis and were fitted with .057 main metering jets.	223 hp at 6500 rpm
Test #11 Standard pistons reinstalled. Edelbrock two-carburetor intake manifold with two Chevy 283 Rochester carburetors. Isky E-2 camshaft and related parts. Spalding two-coil distributor. Hedman headers.	203 hp at 6000 rpm

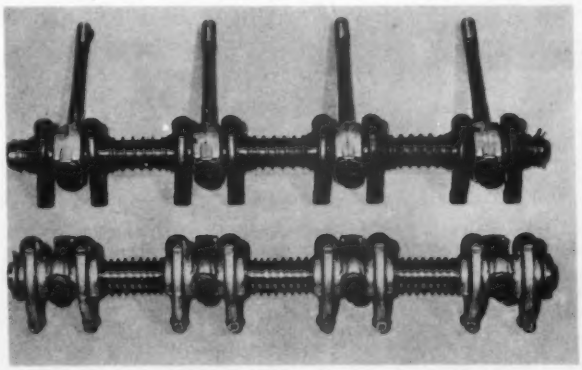
BUICK-TEMPEST V8

Test #1 Engine stock, with several hours of break in time on the dyno. Carburetor main metering jets replaced with jets .002-inch larger. Maximum ignition lead adjusted to 36 degrees. Actual compression ratio was 8.65 to 1.	139 hp at 4500 rpm
Test #2 Olds flat-head pistons installed in place of the Buick recessed-head pistons. This increased the compression ratio to 12 to 1. Otherwise, the engine was the same as it was for Test #1.	156 hp at 5000 rpm
Test #3 Edelbrock two-carburetor intake manifold with two stock carburetors.	177 hp at 5000 rpm
Test #4 Two Chevy 283 two-throat carburetors with .059 main metering jets. Isky E-2 camshaft and related parts. Hedman headers. Spalding two-coil ignition distributor adjusted to provide a maximum of 36 degrees ignition lead.	215 hp at 6000 rpm
Test #5 Carburetor main metering jets were replaced with .062-inch jets. Champion L 63R plugs, which are colder than those used previously.	226 hp at 6000 rpm
Test #6 Modified Isky E-2 camshaft.	230 hp at 6000 rpm
Test #7 Cylinder heads ported. Valve seats enlarged.	233 hp at 6000 rpm and 6500 rpm

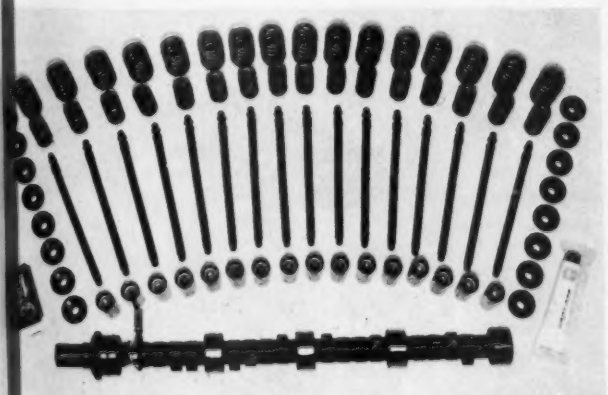
From the left, these pistons are stock Buick, which has recessed head; stock Olds, which has flat head; 12.3 JE for Olds.



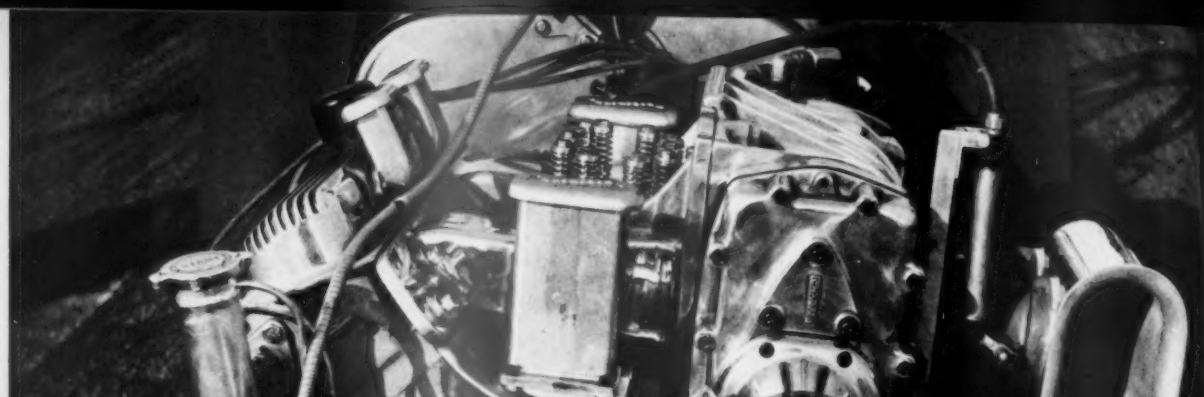
At left is the Edelbrock two-carburetor intake manifold that fits both Olds F-85 and Buick Special. Other is stock F-85.

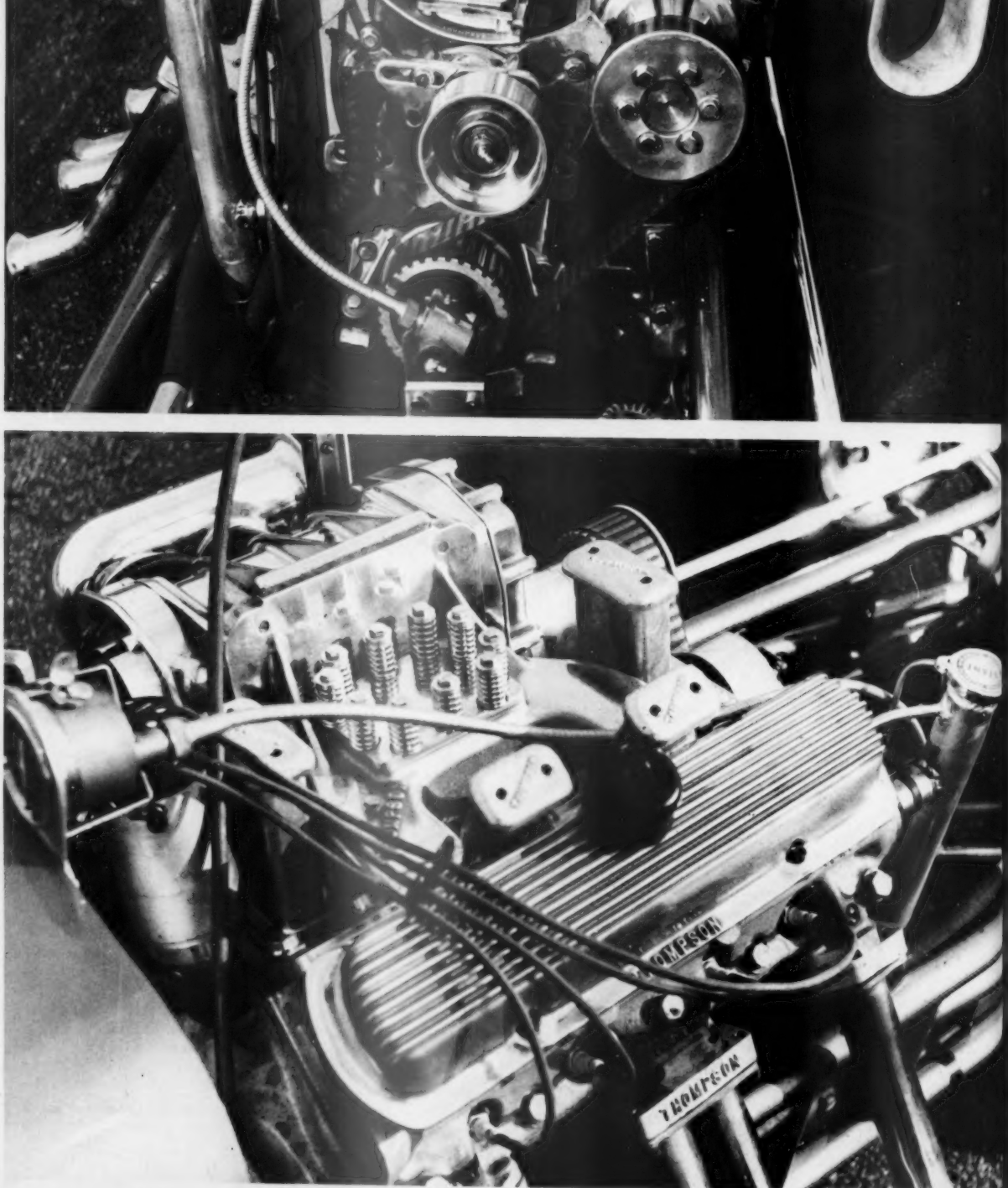


Buick rocker arms, at bottom, are of aluminum alloy. Olds arms, at top, are steel. The arms don't have lash adjusting screws because the engines have hydraulic tappets.



Iskenderian reground camshaft for both F-85 and Buick Special is Full-race type. It is used with special solid tappets. Adjustable pushrods make lash adjustments possible.




THOMPSON

pression ratio and a single-throat carburetor but a more radical camshaft. It is rated at 130 horsepower at 4400 rpm. Torque is 195 pounds-feet. Option number three is the same as number two with the exception of its 10.25 to 1 compression ratio. It is rated at 140 hp. Its torque is 207 pounds-feet.

The strongest of the optional engines is rated at 155 horsepower. It has the 10.25 to 1 compression ratio, the most radical of the three camshafts used in the engines, and a four-throat carburetor. Its torque is rated at 215 pounds-feet at 2800 rpm.

With few exceptions the procedures and special parts for reworking one of the four-bangers are identical to those for the V8's. Boring and stroking limitations are exactly the same, as are porting and other cylinder head reworking operations. The same special pistons, connecting rods, valves, and valve lifters made for the V8's can be used in the four-bangers. Pontiac makes a forged steel crankshaft that can be bought from parts departments of their agencies to replace the standard cast-iron shaft for heavy-duty use. They also have high-performance camshafts, and special shafts in both roller-tappet and flat-tappet grinds are available from camshaft specialty companies. Magnetos, two-coil ignition distributors, and aluminum flywheels are readily available.

For the rodder who wants to install a blower on his Tempest-4, Mickey Thompson Enterprises can supply intake manifolds and complete drive-setups for 4-71 GMC units. Mickey has proved this equipment in his four-banger powered dragster, which has run well over 150 mph at several of the country's drag strips.

CORVAIR

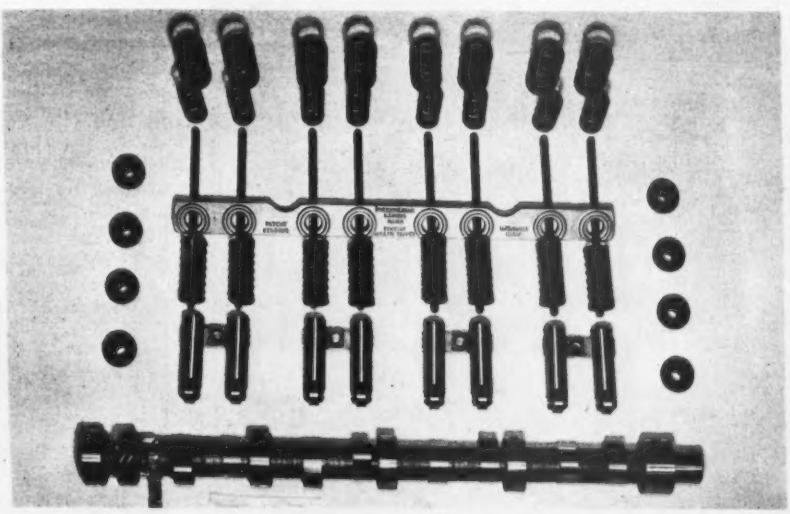
Chevrolet's Corvair differs from all other passenger car engines manufactured in the United States by being air-cooled rather than water-cooled and having its overhead valve cylinders horizontally opposed rather than in an in-line or V configuration. In these details, and because it is mounted at the rear rather than the front of the car, the engine and its placement bear remarkable resemblance to the Volkswagen engine and its placement. The Volkswagen has proved to be the best by far of the small imports. However, the Corvair has six cylinders compared with the Volkswagen's four.

As much aluminum as was practical was used in the Corvair engine to keep its weight low. The handling ability of a car that has its engine in the rear requires that the engine be light. Its crankcase, which is made in two pieces that separate along the crankshaft's centerline, its cylinder heads, which have integral intake manifolds, and several smaller parts are aluminum. The fact that some of its major parts are made of aluminum doesn't have any adverse effect on the engine's reworking potential. However, it does mean that care must be taken when working on the engine so that its aluminum parts, which are not as resistant to rough handling as cast-iron parts, won't be damaged.

Corvair engines for 1960 have a bore of 3.375 inches and a stroke of 2.600 inches. These dimensions give them a displacement of 140 cubic inches. Engines for '61 have 1/16-inch larger cylinders, which makes their bore 3 1/16 inches, that boost their displacement to 145 cubic inches. Horsepower ratings for both '60 and '61 engines are 80 at 4400 rpm. Torque output for the '60 is 125 pounds-feet at 2400 rpm and for the '61 it is 128 pounds-feet at 2300 rpm. Both engines have a compression ratio of 8.00 to 1. All engines have two single-throat Rochester carburetors.

For 1960 an optional 95 horsepower engine was listed and for '61 this option is rated at 98 horsepower. Its compression ratio is 8.00 to 1, the same as for the standard engine, but it has what Chevrolet lists as a high-lift camshaft. Engines in the duded-up Monza chassis that are fitted with automatic transmissions have a compression ratio of 9.20 to 1.

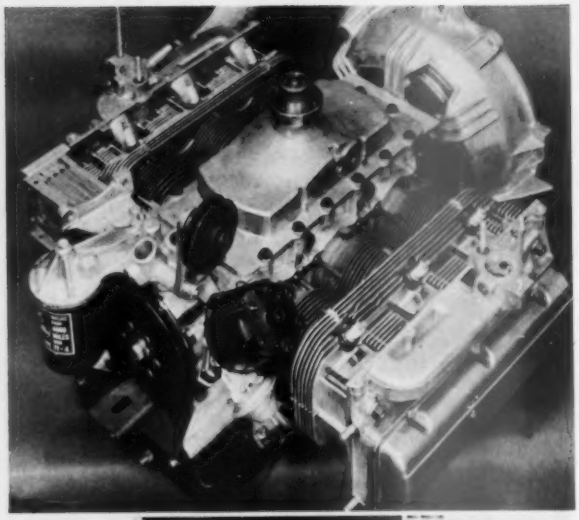
Harry Weber, of Weber Tool Co. in Santa Ana, Calif., was about the first of the hot rod parts manufacturers to dig into the Corvair's reworking possibilities. The results of his efforts were a pair of stroker kits. One kit increases the standard 2.600-inch stroke length .250-inch to 2.850 inches and the other increases the stroke .400-inch to an even 3.000 inches. Pistons in both kits are for 3.500-inch cylinders, which are 1/8-inch larger than standard 1960 cylinders and 1/16-inch larger than standard '61 cylinders. Displacement with a 1/4-inch kit is 164.5 cubic inches and with a .400-inch kit it is 173.0 cubic inches. The displacement increase with the 1/4-inch kit raises the compression ratio to 9.25 to 1; the ratio with the .400-inch kit is 9.65 to 1. These ratios are with standard cylinder heads.



UPPER LEFT—This, too is one of the new compact engines; however, it has been given the Mickey Thompson treatment for go and show. It is a Tempest Four.

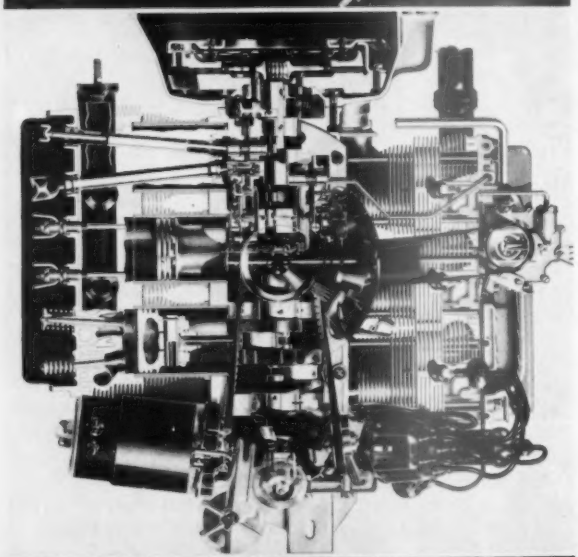
LOWER LEFT—Mickey has Tempests with several bore and stroke dimensions that vary their displacements from the standard 194 inches. The blower is 4-71.

RIGHT—Isky roller-tappet camshaft for Tempest Four. Aluminum plate and extra springs are "Ultra-Rev" kit that add spring pressure to the lifters.



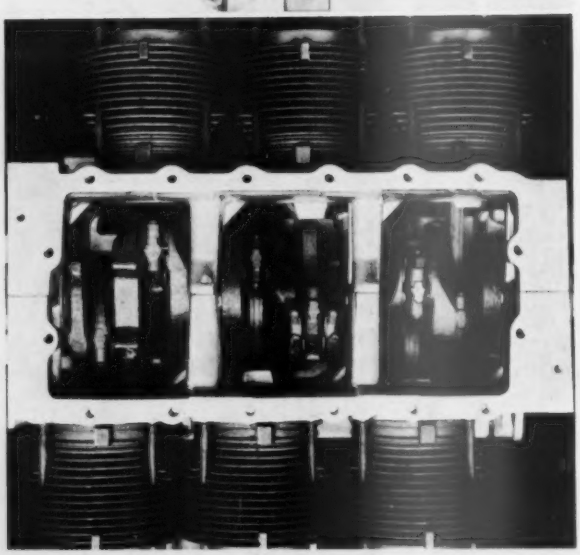
TOP LEFT—Chevy's air-cooled horizontally-opposed aluminum Corvair engine is something different for modern U.S. automobiles.

MIDDLE LEFT—The six-cylinder Corvair functions on conventional principles but most rodders will have to learn how to work on it.



BOTTOM LEFT—Things are a little tight in the Corvair's aluminum crankcase. The case splits down the middle.

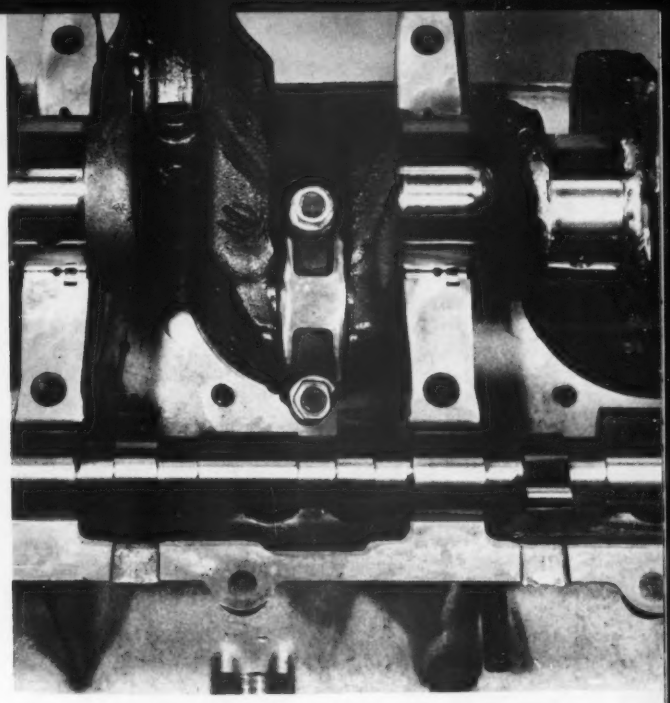
RIGHT—The crankshaft for the Corvair has four main bearings and enough journal overlap to give it adequate strength.



Crankshafts in the kits are stockers that have been stroked by the now-standard welding process. Pistons are Jahns Quality and piston rings are by Hastings. Connecting rods are standard Corvair, as are the main and connecting rod bearings. The rod bearings are standard but the mains are .010-inch undersize. Main bearing journals on the shafts are ground undersize to make their surfaces concentric with the shafts' centerlines and so their diameter will provide the correct clearance in the bearings. All parts in the assembly are balanced and ready for installation.

Also included in the kit is a reground stock camshaft. The reason the camshaft is included is that some of its lobes must be shortened to prevent their being struck by the lower ends of the connecting rods when the rods are rotated by the stroked crankshaft. Material is also removed from the rods to help reduce this interference.

The camshafts in Weber's first kits were for solid lifters but the lifters didn't prove practical because of the expansion characteristics of the basically aluminum engine. When the engine was started and its crankcase, cylinders, and heads expanded as they warmed-up to their normal operating temperatures, the distance from the camshaft to the rocker arms on the heads increased so much more than the total length of the pushrods and the rest of the valve actuating mechanism increased due to expansion that lash clearances between the rocker arms and the valve



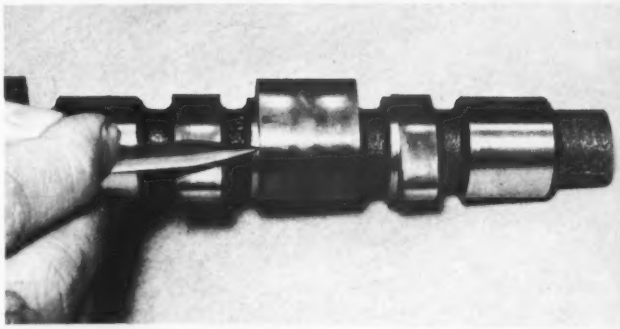
ABOVE—Weber Tool's stroker kits for Corvairs include special Jahns pistons. One of them is shown here, left, with stocker.

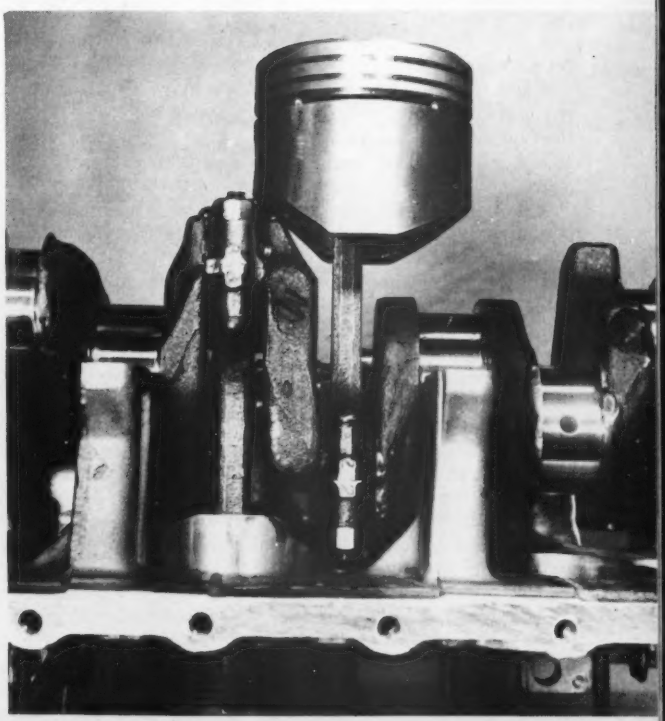
ABOVE RIGHT—The parts in Weber's kits are reworked to provide clearance in the engine for larger arc the rods describe.

RIGHT—Also included in the kits is a reground camshaft. Some of the lobes must be ground on their sides for rod clearance.

LOWER LEFT—Some grinding on the lower ends of the rods is necessary, as indicated with the pencil, to provide clearance.

LOWER RIGHT—The stroke of this shaft is .400-inch longer than stock. Note how the throw side of the piston skirts is cut.





stems became excessive. This caused the valve action to become much noisier than the average driver would tolerate. The most practical solution to this was to use standard hydraulic valve lifters, which automatically compensate for any changes in lash clearances due to expansion of the parts or any other cause. Harry recommends the latest production '61 lifters as they will function correctly at much higher crankshaft speeds than will earlier tappets. Tappets are not included in the stroker kits.

Clearance problems in the crankcase require a bit of trimming here and there in addition to the modifications to the camshaft before the crankshaft and rod assemblies will rotate without hitting something. These problems are less involved with the ¼-inch shaft than they are with the longer one. About all it requires is that the lip on the lower ends of the cylinders be shortened. This can be done by mounting the cylinders in a suitable lathe and machining material from the lips to shorten them approximately .080-inch.

For the .400-inch shaft the barrels must be shortened .125-inch. Also, clearance areas must be ground in the inner surfaces of the crankcase for each of the connecting rods. The area for a specific rod is in the crankcase's surface directly opposite the cylinder in which the rod moves. In other words, the clearance is for the lower end of the rod when the rod's piston is at bottom center in its cylinder.

One of Weber's stroking kits really helps a Corvair's performance at all engine and car speeds but the results would be considerably better if the engine's carburetor venturi area were also increased. The area provided by two standard carburetors is barely adequate for an engine of standard displacement. When Weber's kit first became available there wasn't much a fellow could do about this because of the integral intake manifolds used on the heads but now a very simple and practical solution is available from Bill Thomas, whose address is P.O. Box 4127, Catalina Station, Pasadena, Calif.

Thomas has designed and made available a kit of parts that makes doubling a Corvair's venturi area by installing two additional stock carburetors an extremely simple matter. Results of performance tests with a standard 95 hp Corvair on drag strips around Los Angeles were approximately 71 mph in 18.50 to 19.00 seconds with the stock carburetion setup and approximately 85 mph in 16.14 seconds with the four-carburetor setup. The car used for these tests was stock with the exception of Corvette valve springs, which are stiffer than standard Corvair springs and allow the engine to turn a little tighter before the valve lifters start holding the valves open, and an open exhaust system. It has a three-speed transmission and 3.89 to 1 rear axle gears.

Although the four-carburetor setup helps tremendously on a drag strip, the best things about it are the way it improves acceleration at normal highway cruising speeds and the improvement it makes in a car's hill climbing ability. Also, the simple modification of mounting all the carburetors with their float bowls forward eliminates carburetor flooding in turns. Still another improvement is that the setup eliminates all the flat spots during acceleration normal to Corvair's.

The basic parts of Thomas's kit are a pair of cast-aluminum adaptor blocks that support the two additional carburetors and a pair of flat aluminum adaptor plates that enable the original carburetors to be rotated to place their float bowls forward. All parts necessary for installation of these adaptors are included in the kit. The installation is extremely simple and is done without removing the cylinder heads from the engine.

Securing the adaptor flanges for the standard carburetors to the manifolds is merely a matter of fastening them to the manifold's flanges with flat-head screws after the carburetors and their studs have been removed. The carburetors are then installed on the adaptors.

Mounting the adaptor blocks for the two extra carburetors is a little more involved but still simple. After locating a block on the manifold according to instructions, two holes are drilled through the top of the manifold and threaded for ¼-inch, 20-thread Allen socket-head capscrews that secure it to the manifold. The block is then bolted in place and a new port is cut in the top of the manifold with a hole saw supplied in the kit. The saw is inserted through the opening in the block. Using the block as a guide in this manner guarantees that the port will be in the correct location in the manifold. A ¼-inch electric drill does nicely for rotating the saw; however, the saw could be rotated by hand with a hand drill or brace of some sort if an electric drill weren't available.

After the port has been cut, the block is removed and the aluminum chips from the drilling and sawing are cleaned out of the manifold with a vacuum cleaner of some sort. The port is large enough to allow the hoses used for most vacuum cleaner attachments to be inserted in it. The block is then re-installed but this time the joint between it and the manifold is filled with a bonding agent called "Devcon," which is supplied in the kit. Devcon bonds the two members together so tightly that the joint between them is as strong as the aluminum. It prevents air leaks between the parts by effectively sealing the joint.

All that remains to be done is to install the two additional carburetors, their fuel lines, and the throttle linkage. Installation of throttle linkage requires cutting the ends from the standard arms on the carburetor and bolting special arms to the portions that remain.

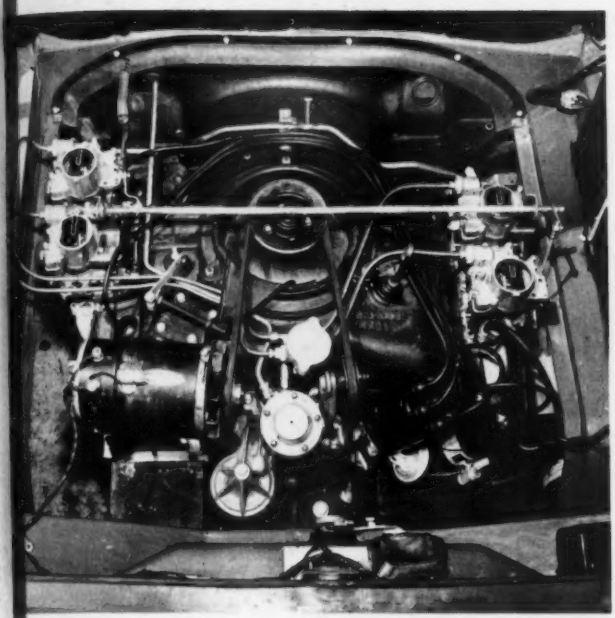
Claims for Thomas's four-carburetor setup aren't exaggerated. It hasn't been tried yet on a bored and stroked engine but it really makes a standard displacement engine come to life. The faster the engine turns, the better it feels. Mike Jones, who is working with Thomas on the development of the kit and does much of the test driving, says that gasoline mileage, for drag racing, driving to and from the drag strip, and buzzing from place to place in normal city traffic and on freeways, is an honest twenty miles per gallon. This isn't bad because Mike likes to use the extra performance the carburetor setup makes possible.

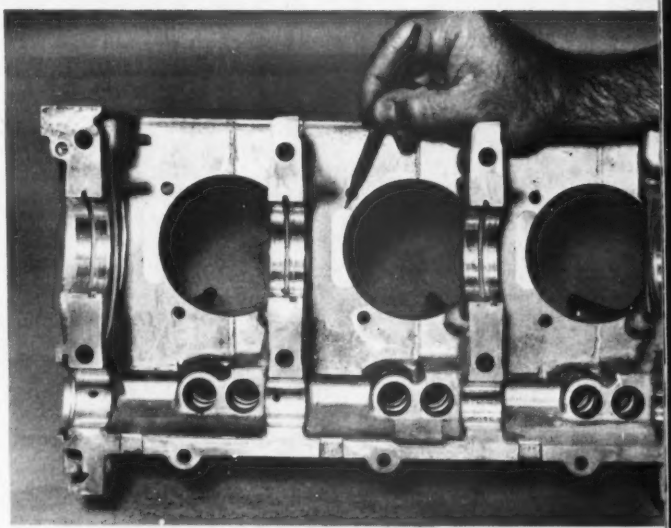
Thomas has also worked out a series of modifications for Corvair cylinder heads. Complete instructions and blueprints for this job, which requires considerable machine work, are available from Thomas. Included are instructions for making and installing oversize valve seat inserts for modified Corvette intake and exhaust valves, installing Corvette valve springs, and replacing the standard rocker arms with 1.75 to 1 ratio 348 Chevy arms. These modifications are said to make a considerable improvement in a Corvair's breathing ability.

VALIANT-LANCER

Chrysler Corporation's Valiant was another six-cylinder addition to the 1960 compact field. Its cylinders followed conventional practice by being in line but its cylinder block was inclined at a 30-degree angle toward the right side of the car's engine compartment. The advantages of rotating the cylinder block from the vertical in this manner are debatable but it gives an advertising department something to talk about and it doesn't have any apparent disadvantages.

The '60 Valiant's 170 cubic inches of displacement were provided by 3.4-inch cylinder bores and a stroke of 3.125 inches. The compression ratio was 8.2 to 1. Horsepower output was rated at 101 at 4400 rpm. Maximum torque was 155 pounds-feet at 2400 rpm. Its overhead valves were actuated by conventional rocker arms and solid valve lifters. The cylinder head had an intake port for each cylinder and





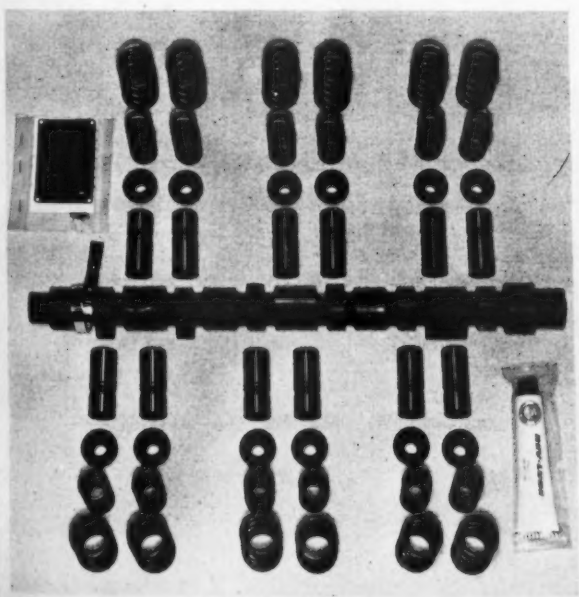
ABOVE—One of the latest improvements for Corvair engines is the excellent four-carburetor setup available from Bill Thomas.

ABOVE RIGHT—This aluminum adaptor flange and adaptor block are basic parts of Thomas' carburetor installation kit.

RIGHT—Installation of Weber's stroked shafts requires clearance notches in the crankcase for the lower ends of the rods.

LOWER LEFT—Isky also has a camshaft kit for Corvairs. Solid tappets can be problem in Corvairs due to expansion troubles.

LOWER RIGHT—Synchronizing gauge, such as this "Uni-Syn," is recommended for adjusting idle of engines with four carbs.



the carburetion setup consisted of a single-throat carburetor on a long-legged manifold.

In 1961 Chrysler Corporation retained the Valiant but also added the Lancer to their compact offsprings. The 1960 Valiant engine, unchanged from its '60 specifications, became the standard power plant for both cars; however, a 225 cubic inch engine that has the same external appearance as the 170-incher was offered as an option in the Lancer.

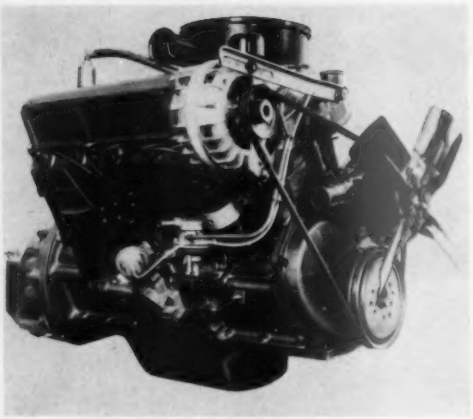
The 225 has the same bore diameter as the Valiant but its stroke is a full inch longer, which makes it 4⅛ inches. In these days this is a long stroke for a production engine. Its cylinder block differs from the 170 block by being taller to help compensate for the longer stroke. Also, its connecting rods are longer. It has the Valiant's 8.2 to 1 compression ratio and cylinder head. Its horsepower is rated at 145 at 4000 rpm. Maximum torque is 215 pounds-feet at 2800 rpm.

A second option for the Lancer is another 225 cubic inch engine identical to the standard 225 except that it has an aluminum cylinder block. The aluminum block effects a reduction of approximately 66 pounds in the engine's weight.

In 1960 a kit of high-performance parts was listed by the factory for the Valiant. Included in the kit were a four-throat carburetor on a "ram-tuned" manifold, a camshaft that provided more radical valve action, pistons that boosted the compression ratio to 10.5 to 1, and an exhaust manifold that was nothing more than cast-iron headers. As none of these parts were installed at the factory there's no doubt their principal purpose was to make a few really hot engines available for Daytona and a few other similar competition events. This little bit of strategy paid off well at Daytona when the Valiants blew off the other compact entries by several miles per hour.

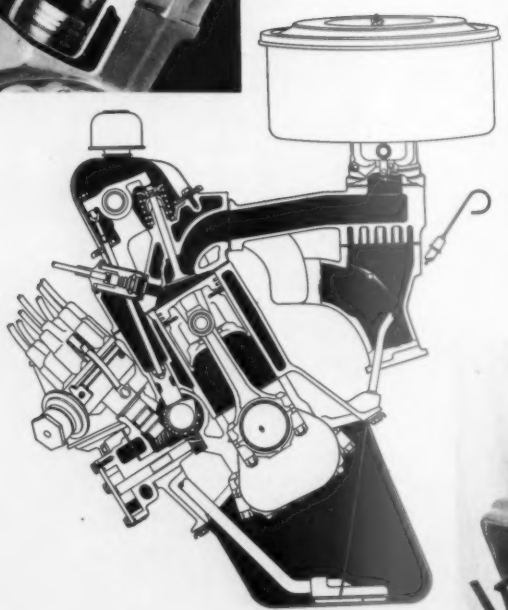
Actually, Lancer and Valiant owners don't seem to be hot rodder types. If there are any radically modified engines of these makes around, their owners have succeeded in keeping the news to themselves. Among the special equipment advertised for them are aluminum flywheels, two-carburetor intake manifolds, and reground camshafts and kits. Also available are the "regular" special parts, such as two-coil distributors, Vertex mags, racing pistons, etc.

Because the same cylinder head is used on them, the

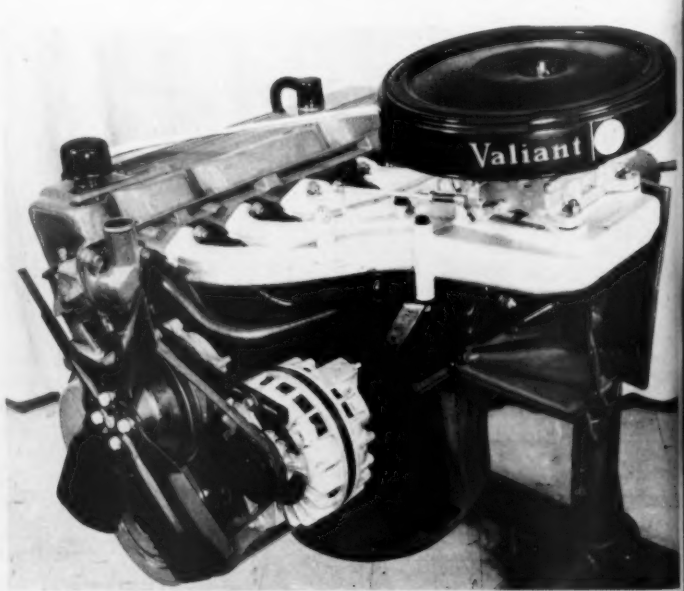


Chrysler Corp.'s Lancer-Valiant engine is made in three versions. One has an aluminum cylinder block.

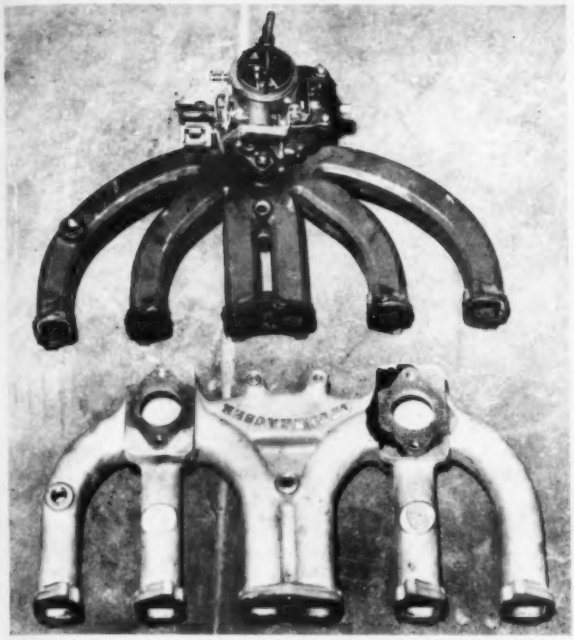
The larger version of the Lancer-Valiant is also used in Dodge Darts. The rocker arms are stamped steel.



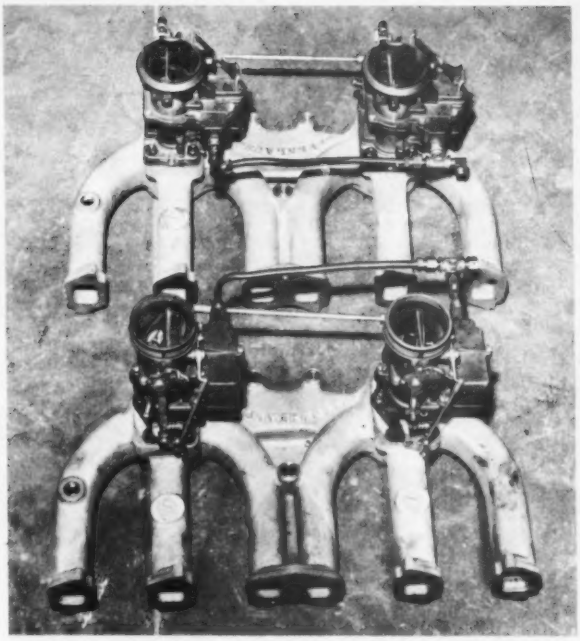
Lancer-Valiant engines are of conventional construction with the exception of being angled to the right 30 degrees from the vertical.



Valiant engine fitted with factory "Hyper-Pak" high-performance kit. Long-legged aluminum inertia-ram manifold supports a four-throat carb.



Better performance from Lancer-Valiant can be gained through the addition of another stock carburetor on a special manifold.

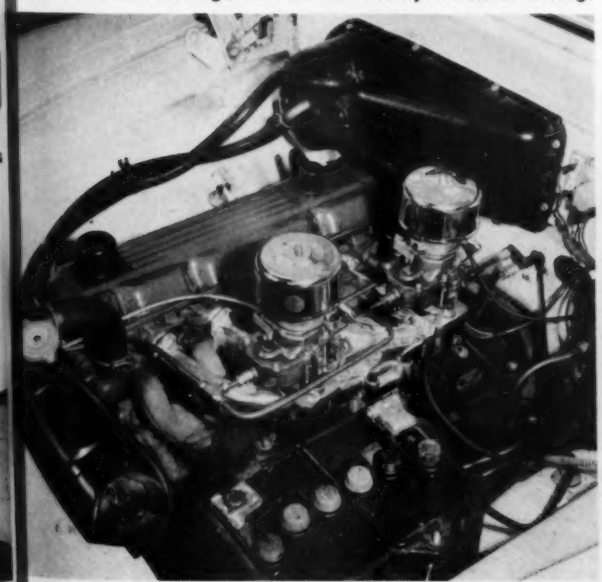


Dual manifolds by Offenhauser Equipment Corp. for Lancer-Valiants accommodate carburetors of several different types.

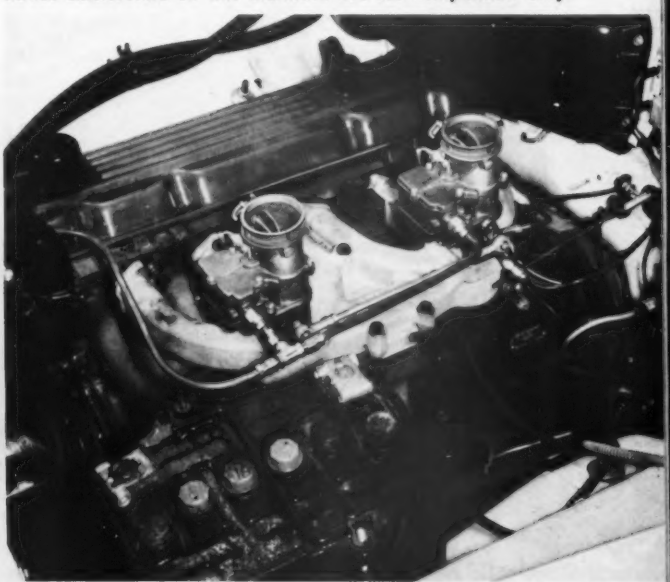
same two-carburetor manifolds fit both the 170 and 225 engines. Typical of these are the manifolds manufactured by Offenhauser Equipment Co. Offenhauser's setup is available for two stock single-throat carburetors and single-throat carburetors of other types that have either three or four-bolt flanges, and for Stromberg 97 two-throat carburetors. It is made of aluminum and has long passages similar to those in a standard manifold. However, because the passages are in two sets of three rather than one set of six, they are more nearly equal in length. The two sets of passages are connected with a balance passage that joins them below their carburetor risers. The manifolds are exhaust heated.

Single-throat carburetors are mounted on the manifold with their bowls forward but Stromberg 97's are mounted with their bowls at the side so their throats and throttle valve action will provide more even distribution of fuel and air mixture to the manifold's passages. Single-throat carburetors are recommended for normal driving and the two-throats, because of their greater venturi area, are recommended for competition engines only. **DON FRANCISCO**

The standard carburetors used for this dual installation are recommended for engines to be used mostly in normal driving.



Because of the venturi area they provide, the Stromberg two-throat carburetors on this manifold are for competition only.



WHAT'S NEW IN TRANSMISSIONS

CHRYSLER CORPORATION

AS FELLOWS who like synchromesh transmissions for competition and street rods are well aware, big, husky transmissions are difficult to find these days. Cad-LaSalle boxes, which were the best and the most popular of the strong transmissions for competition cars, and Packard and Buick units have just about disappeared from the country's wrecking yards. And whenever a fellow does find one of them he usually can't afford it because of its inflated price.

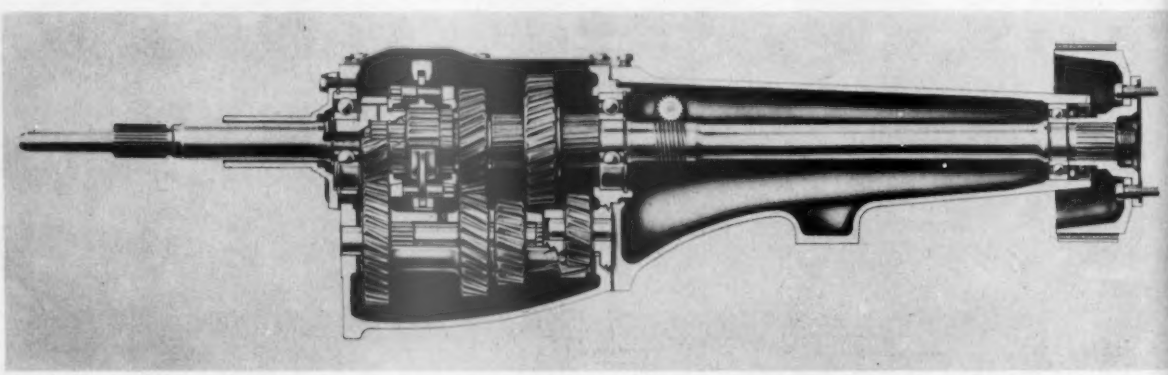
The shortage of suitable synchromesh boxes for hot rods is the result of nothing more than an inadequate supply for the demand. Production of the popular transmissions was stopped many years ago. It was inevitable that their high mortality rate on drag strips and at the hands of speed-shifting street drivers would soon bring the bottom of the barrel into sight. But now there is new hope for the stick-shift fraternity: The Chrysler Corporation has begun production of a new heavy-duty synchromesh transmission for their 1961 high-performance automobiles that looks as though it could be easily adapted for hot rod use.

Development work on synchromesh transmissions was at what amounted to a stand-still at Chrysler Corporation, and this probably also applied to Ford and General Motors, during the many years when sales statistics gave the impression automatic transmissions were going to eliminate the manually-shifted boxes altogether. But after sales of cars equipped with synchromesh boxes reached their low in '57 and then started a climb back that has lifted them higher in sales percentages each year, Chrysler decided to design and manufacture an improved series of synchromesh transmissions that would fulfill the demands for their entire line.

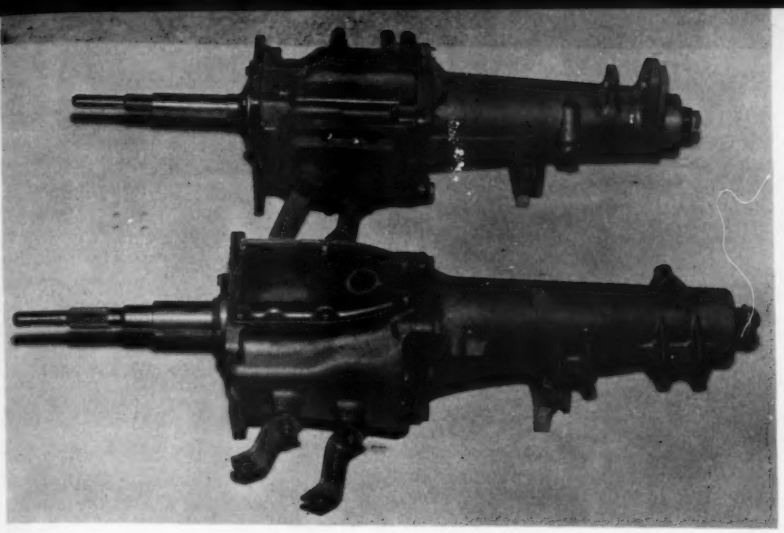
The steadily growing group of persons who buy cars equipped with synchromesh transmissions consists, primarily, of motorists who are interested more in economy than in driving convenience. The lower first cost of manually-shifted transmissions and their better fuel mileage potential make them especially attractive to buyers of compact cars. This is proved by the large number of compacts sold with them. A minor percentage of crash box purchasers are the romanticists who take delight in shifting their own gears, as their grandpappy did in his Stutz Bearcat, and fellows who plan to use their car for dragging or track racing.

Chrysler's engineers realized when they started their transmission development program in '58 that transmissions of two sizes would be necessary. One would accommodate the Corporation's small engines and the other would be for the larger V8's. It is the one for the larger engines, which Chrysler calls the heavy-duty model, in which we are interested. It is strong enough to hold the torque and power outputs of the Corporation's 413 cubic inch V8 that is standard in Chrysler's and Imperial's and available as a high-performance option in nearly all the Corporation's full-size cars.

Before 1961 Chrysler used synchromesh transmissions designed and manufactured by the Warner Gear Co. A visual comparison of the internal parts of their new heavy-duty box, one of Warner Gear Co.'s equivalent transmissions, and a Cad-LaSalle box show all three to be approximately equal in gear and shaft sizes and, presumably, in strength. However, construction details of the new Chrysler box differ considerably from those of the Warner box. Most of these differences are the results of Chrysler's effort to



Chrysler Corporation's new heavy-duty synchromesh transmission is a sturdy, well-designed unit that may find a place in rodding.



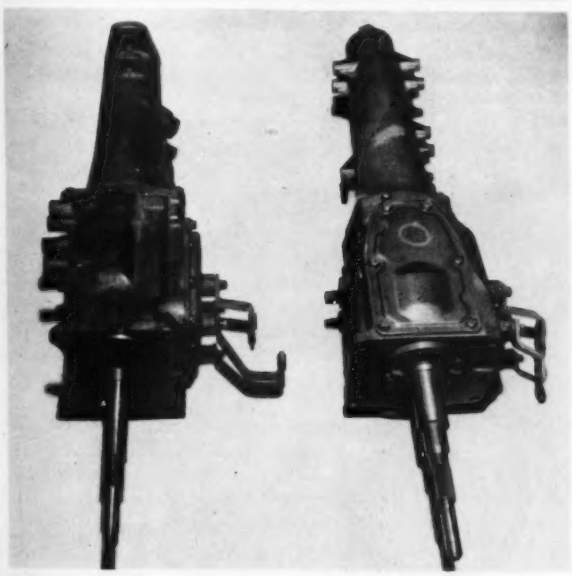
LEFT—Chrysler's new heavy-duty trans supercedes the Warner box above it. Camera angle makes the Warner box appear to be shorter than it is.

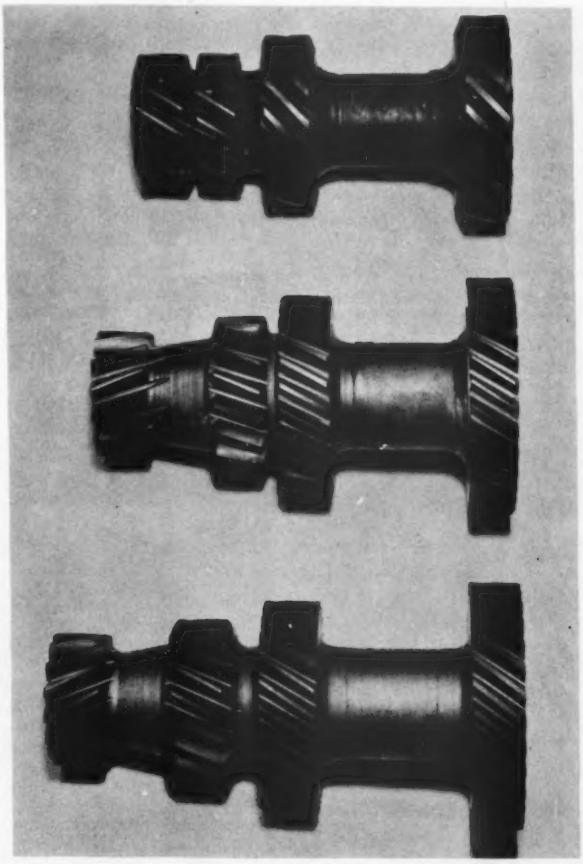
LOWER LEFT—Its "closed case" design, and careful consideration of details, enable the Chrysler box, at right, to be 13 pounds lighter than the Warner box.

BELOW—These countershaft, or "cluster," gears are, from the top, Cad-LaSalle, Chrysler heavy-duty, and Warner. All appear to be quite husky.

Transmission Specification Chart

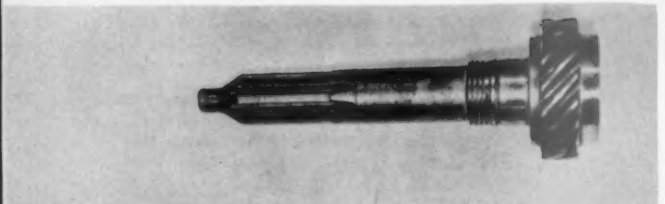
Specification	Transmission	
	Chrysler Heavy-duty	Warner Gear Co.
Overall length	37 inches	36 1/8
Mainshaft projection	10 7/8	10 3/8
Case length	8 15/16	9 1/2
Rear housing length	15 3/4	15
Center of motormount pad to front of case	15 7/8	15 7/8
Weight	81 pounds	94 pounds





Transmission Gear Ratio Chart

Transmission	Low	Second	Ratio drop from Low to Second	Ratio drop from Second to High
Chrysler Heavy-duty	2.55	1.49	1.06	1.49
Warner Gear Co.	2.49	1.58	.91	1.58
Cad-LaSalle	2.39	1.52	.87	1.52



ABOVE—These clutch shaft gears are, from the top, Cad-La Salle, Chrysler, Warner. Angle of teeth on Cad gear is greater.

make the new transmission as compact, light in weight, and reasonable in cost as possible.

One difference in the transmissions that is immediately apparent is the design of their cases. Shift levers for both are on the left side of the case but the shafts for the Warner box's levers are supported by a cover that bolts to the case whereas the Chrysler shafts are supported by bosses cast as an intergal part of the case. The access opening in the Chrysler case, through which the transmission's internal parts are installed, is sealed by a light weight stamped-steel cover. According to Chrysler, designing the case as they did, which they call "closed-case" construction, makes it so much stiffer that its walls can be .020-inch thinner than those in the Warner case.

Another design difference is that the ball bearing that supports the mainshaft in the case is pressed into the case rather than into the tailshaft housing as it is in the Warner transmission. This construction not only saves weight but it also eliminates an extra machining operation and another joint in the assembly that could affect the mainshaft's alignment. This bearing also resists end thrust exerted on the mainshaft by the propeller shaft assembly. In the Warner box the bearing in the end of the extension housing, which supports the propeller shaft end of the mainshaft, takes this thrust.

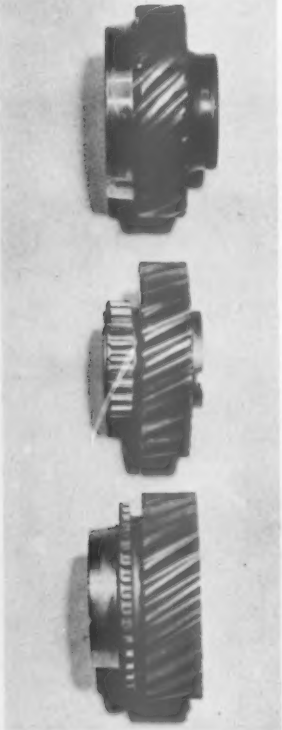
Heavy-duty ball bearings support the Chrysler transmission's clutch gear shaft and the mainshaft at the rear of the transmission housing and at the rear of the tailshaft

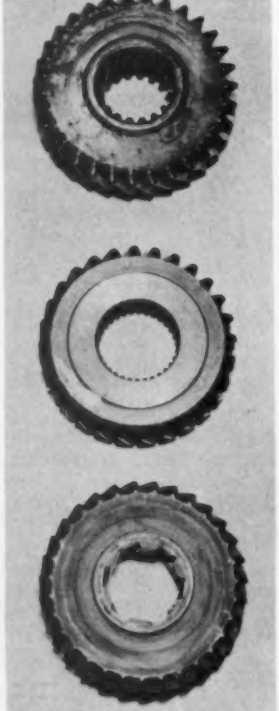
LOWER LEFT—Dimensions of the second gears for the 3 boxes are similar; however, each gear has a different number of teeth.

LOWER RIGHT—Widths of the second gears vary slightly, with Warner's the widest and the Cad and Chrysler about the same.

LOWER LEFT—Low and reverse sliding gears are also similar in size. The Chrysler gear has three less teeth than the others.

LOWER RIGHT—The Chrysler gear, in the middle, does not have a groove for a shifting fork. It's moved by a U-shaped member.





housing. Large-diameter roller bearings support the front end of the mainshaft in the clutch shaft and two rows of needle bearings support each end of the cluster gear. Splines on the mainshaft, on which the low and reverse sliding gear moves, are of the involute rather than the straight type. The reason for this change is said to be that they allow the gear to slide easier when shifts are made than when straight splines are used.

Because of the spacing of the mounting bolt holes on their front flanges, the Chrysler and Warner transmissions require different bell housings. Also, the piloting flange on the front bearing retainer is smaller on the Chrysler box than it is on the Warner. Clutch shaft splines for both transmissions have an outside diameter of 1-inch but the Chrysler shaft has 23 small splines in place of the 10 larger splines on the Warner shaft. The throwout bearing pilot on the Chrysler transmission is 1.638-inch in diameter; on the Warner box its diameter is 1.25-inch. Tailshaft splines, for the universal joint, are the same on both transmissions. They are approximately 1.220-inch in diameter and have 23 splines. The mounting flange on the front end of the Chrysler case is ⅝-inch thick; on the Warner case it is approximately ½-inch thick. Other specifications for the transmissions can be found in the accompanying charts.

A definite improvement in the Chrysler transmission for rough use is its pin-type, or "blocking-type," synchronizer. This unit is similar to the one in Cad-LaSalle transmissions but altogether different from the more conventional and less sturdy synchronizer in Chrysler's former transmission and in most transmissions of other makes.

As many of today's hod rodders have grown up with automatic transmissions, they may not be familiar with the purpose of a synchromesh transmission's synchronizer and the way it operates. A description of the construction and operation of Chrysler's synchronizer should serve the double purpose of introducing this essential transmission part to those unfamiliar with it while at the same time familiarizing rodders with one of the Chrysler transmission's better features.

Gears in synchromesh transmissions differ from those in the early, now obsolete, transmissions that didn't have synchronizers by being engaged at all times with the gears that drive them. In early transmissions the gears were meshed with the gears that rotated them only when they were being used. This meant that a gear had to be moved on its shaft to engage its teeth with the driving gear whenever that particular transmission ratio was to be used. In a synchromesh box the gears are brought into use by merely locking them to the mainshaft with the synchronizer.

Only second and high speeds in modern three-speed transmissions are synchromesh gears. Low gear is still set up as it was originally; it must be moved on its shaft and engaged with the gear that drives it. In some of the special four-speed transmissions now being built all forward speeds are synchromesh.

The heart of the pin-type synchronizer in Chrysler's new transmission is a hub and shift sleeve assembly. The hub has inner splines to match those on the mainshaft and outer splines that match those in the shift sleeve. This assembly is between the main drive gear on the clutch shaft and the second gear, which is free to rotate on the mainshaft. It rotates the mainshaft. The outer circumference of the shift sleeve is reduced in width to form a narrow flange in which there are six equally spaced holes. The outer portion of this flange rotates in the high and second gear shifting fork.

On the rear end of the main drive gear and the front end of the second gear are splines that match those on the outer diameter of the synchronizer's hub and the inner diameter of the shift sleeve. When the shift sleeve is moved forward, one end of its splines slides over the splines on the main drive gear. This locks the main drive gear to the synchronizer's hub, which is the same as locking it to the mainshaft. This places the transmission in high gear.

When the sleeve is moved rearward, it slides over the spline on the second gear. This locks the second gear to the mainshaft to place the transmission in second speed. Torque from the main drive gear is transmitted to the cluster gear, from the cluster gear to the second gear, from the second gear to the synchronizer, and from the synchronizer to the mainshaft.

Because the main drive gear and the mainshaft are rotating at different speeds when shifts are made it would be difficult to slide the shift sleeve onto the spline of the desired gear unless some means of equalizing the speeds of the parts were provided. This is accomplished by a pair of cone clutches that consist of two members for each of the gears engaged by the synchronizer. One of the members is a bronze ring with a tapered outer face that rotates with the gear. The other is a steel ring with a tapered inner face that rotates with the shift sleeve. The connection between the ring and the sleeve's flange is three pins attached to the ring. These pins have two diameters. The smaller diameter is a loose fit in the holes in the flange.

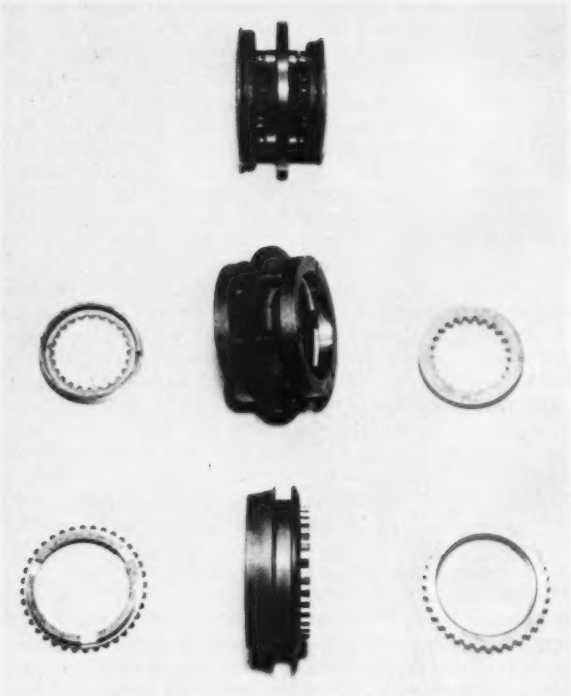
When the taper in the steel ring is forced onto the corresponding taper on the outer circumference of the bronze ring, a drag, or braking action, is created between the two members. This drag quickly equalizes the speed of the gear being engaged with that of the mainshaft, which is being rotated by the car's rear wheels and driveshaft, by increasing or decreasing the gear's speed.

Pressure to force the steel ring onto the bronze ring originates in the shift sleeve. It is transmitted from the sleeve's flange to the ring by the step between the two diameters on the pins that rotate the ring. Although the smaller diameter of the pins is a loose fit in the holes in the sleeve's flange, the larger diameter is a snug fit and aligns with the holes only when the speeds of the steel and bronze rings are the same. The spline in the shift sleeve can't slide onto the gear until the holes in the sleeve's flange slip over the larger diameter of the pins.

Misalignment between the pins and the holes in the shift sleeve's flange that results when the speeds of the parts aren't synchronized and the pins are trying to rotate the sleeve, or the sleeve is trying to rotate the ring and pin assembly, causes the sleeve to exert pressure on the steel ring. As soon as the resultant drag equalizes the speeds of the parts, the sleeve slips over the pins and its splines slide onto the splines of the gear being engaged.

To enable the fast and violent shifts necessary for drag racing and other types of driving that require maximum performance from the car to be made, a slight chamfer is provided on the larger diameter of the step between the two diameters on the pins and on the ends of the holes in the shift sleeve flange. These chamfers are narrow enough to prevent their interfering with the ability of the flange to exert sufficient pressure on the ring for normal shifts but when enough force is exerted on the flange by the car's shift lever they allow it to slip over the pins although the speeds of the parts aren't exactly the same.

As can be seen on the accompanying chart, the ratios of the gears in Chrysler's new heavy-duty transmission aren't as ideal for drag racing as the ratios in Cad-LaSalle boxes. The variation between the ratios for second gear is slight but for low gear it is somewhat more pronounced. It must be remembered that differences between the ratios will be magnified by the ratio of the car's rear axle gears. With



The Cad-LaSalle synchromesh unit, top, and Chrysler's, middle, are of a different type than Warner's and better for hard use.

axle gears that have a ratio of 4.00 to 1, for instance, the final low gear ratio with the Chrysler box would be 10.20 to 1 and with the Cad-LaSalle box it would be 9.56 to 1. But, even though they do vary the wrong way from Cad-LaSalle ratios, the ratios are still within the acceptable bracket for a high-performance car. Just wind the engine a little tighter in low to compensate for the larger drop in engine rpm to that wild 1.49 second gear. The ratio drop from second to high is so close to that of a Cad-LaSalle that the average fellow wouldn't notice the difference.

Used in combination with the heavy-duty Chrysler box is an 11-inch clutch disc that has 123.7 square inches of friction area. This should eliminate any problem a fellow might otherwise encounter in trying to find a large enough driven disc with a hub to match the spline on the transmission's clutch shaft.

BEEFING THE FALCON-COMET AUTOMATIC TRANSMISSION

Automatic transmissions are usually designed with just enough torque transmitting capacity, plus a reasonable safety margin, for the engines on which they will be used. The principal reason for this as far as automobile manufacturers are concerned is that a transmission's cost goes up as its torque capacity becomes greater.

The Ford Motor Co. stuck to the white line down the middle of the road with their transmission for Falcon-Comet cars by using the guts of a unit already in production instead of designing a completely new smaller transmission, and reducing the transmission's cost a few pennies by installing a weaker clutch assembly in it. The guts they used are from their '59-'60 Ford-O-Matic two-speed trans. This unit was adapted to the smaller engines by fitting it into a new housing.

Using the internal parts of their standard automatic trans for the little engines was the nicest thing Ford engineers could have done for hot rodders who like the little cars. Changing the clutch assembly reduced the torque capacity of Falcon-Comet units but it's a simple matter to replace the clutch with one designed to hold Ford's V8's.

All that is involved in the clutch beefing is removing the clutch's two plates and two discs and the servo piston that transmits pressure to them and replacing them with a standard set of five plates and five discs and a servo piston for a big transmission. Ford numbers for these parts are 77573 for the plates, 7B164 for the discs, and CA COAP 7A262-C for the piston.

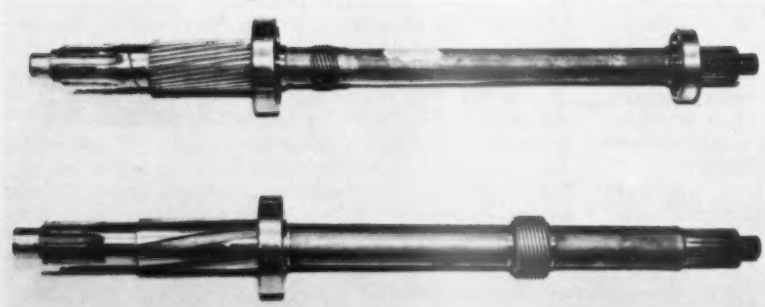
The only difference between the servo pistons for the two transmissions is that the one in the little unit is thicker to compensate for the thinner disc and plate assembly. If a piston for a big transmission isn't available, the original piston can be used by reducing its overall thickness to .447-inch. Any good machine shop should be able to do this but the job must be done carefully so the piston's thickness will be uniform around its circumference. Material removed from a piston for this purpose is machined from its open side, not the side that is next to the clutch assembly.

As automatic transmission work requires a thorough knowledge of the transmission involved and special tools and gauges to enable it to be disassembled and assembled correctly, a hot rodder who doesn't qualify as a transmission specialist should not try to rebuild or beef his own automatic. Chances are that if he does his transmission will become a "basket case" that only a transmission specialist will be able to salvage.

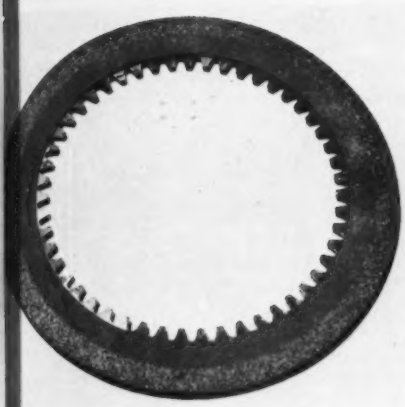
BEEFING PONTIAC'S NEW THREE-SPEED HYDRA-MATIC

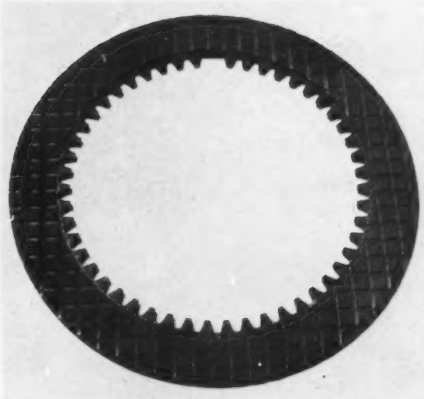
Prior to 1961 all Hydra-Matic transmissions had four forward speeds. Now, the only four-speed Hydra-Matics manufactured are used in Cadillacs and Star Chief and Bonneville Pontiacs. Those for Catalina and Ventura Pontiacs and all Oldsmobiles have only three speeds.

In addition to being less complicated internally than its predecessors because it has only three speeds, the new



Chrysler's mainshaft, above, is about the same length as Warner's but it differs by having involute rather than straight splines. The better production control possible with involute splines is said to make a better-shifting trans.

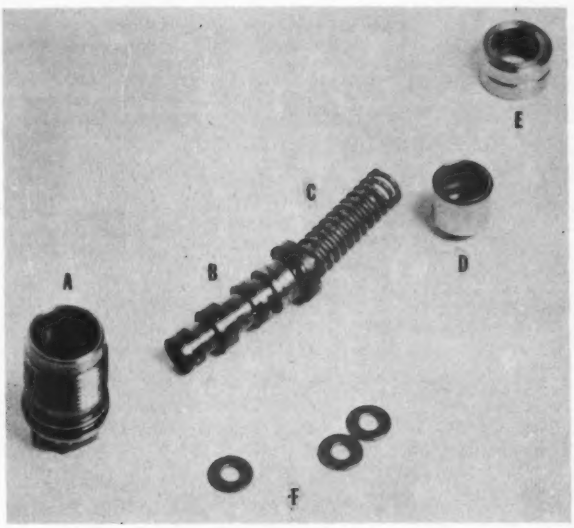




ABOVE—This is standard Hydra-Matic clutch plate. It is o.k. for use with stock engines but not with reworked power plants.

ABOVE RIGHT—Special non-metallic facing on this disc does good job in some B&M Automotive Hydro-Stick transmissions.

ABOVE FAR RIGHT—B&M uses metallic-type facing in Hydro-Stick transmissions they build for engines fitted with blowers.



This is an exploded view of the parts that form the pressure regulator valve in Pontiac's new three-speed Hydra-Matic. The parts are (A) valve housing; (B) valve; (C) pressure control spring; (D) piston; (E) spacer; (F) three flat steel washers that are used to increase the spring's pressure. Increasing the spring pressure allows the trans to be held in second gear until the driver shifts to high manually with the shift lever.

transmission differs in still another way by having an aluminum alloy rather than cast iron case and bellhousing. The internal simplification and the aluminum alloy parts reduce the new transmission's weight considerably; however, as sometimes happens with parts of this type, the weight reduction effected by the switch from cast iron to aluminum was accompanied by a loss in case and bellhousing strength.

The new case and bellhousing are strong enough for normal driving but where their weakness seems to show itself is during "stall" checks. A stall check is made by holding the car with its brakes so it can't roll and then opening the throttle with the transmission in gear. This applies engine torque to the transmission and the rest of the car's driveline. If enough torque is applied in this manner its reaction can become great enough to twist the case or bellhousing. This means that it isn't a good idea to hold a car with one of these transmissions on the starting line of a drag strip with the brakes while revving its engine for a jump start. If the engine is turned tight enough it's entirely possible that the bellhousing or transmission case will be damaged. This problem doesn't exist when the car's rear wheels are free to rotate.

The new three-speed box was tried for drag racing by a group of Pontiac fans in the east and found lacking. Low gear was great. With the shift lever in Low position the trans stays in low regardless of engine or car speed. Second gear was a different proposition. When the shift lever is moved from Low to R, which is the second speed position, with the engine turning the 5700 rpm necessary for good drag strip performance, the trans merely pauses in second and then drops into third. This doesn't allow engine speed to climb back up to 5700 rpm, where it should be, before the shift is made to high gear.

After a bit of experimentation it was found that boosting the oil pressure in the transmission's circuits enables the trans to stay in second gear until engine rpm's climb to a point close enough to the desired speed to make the shift to high gear at a practical rpm for good performance.

Oil pressure in the transmission's shifting mechanism is controlled by a relief valve that opens when a pre-determined pressure is reached. The simplest method of raising the pressure is by placing a spacer approximately 3/16-inch thick between the spring's end and the surface on which it seats in the valve's piston. Three flat washers of normal thickness were found to have the correct total thickness for the spacer.

With the pressure boosted in this manner the maximum speed the engine will turn before the transmission shifts automatically to high gear depends on the transmission's

internal condition. If the trans is tight internally so the pressure can't escape, it will remain in gear longer than it will if it has many places where oil can escape and drop the pressure. When the maximum rpm is reached, the trans will shift automatically to third gear. The shift can be made manually before this engine speed is reached by moving the shift lever.

An adjustment that must be made in conjunction with boosting the pressure involves the linkage from the throttle linkage that actuates the pressure control lever on the transmission's side. This linkage should be adjusted to shorten the time required for the second-to-high-speed shift to the minimum.

Boosting the transmission's oil pressure and adjusting its throttle control lever for the best shifts doesn't destroy its action for normal driving. With the shift lever in DR position the transmission shifts automatically, just as it did before, but the shifts are more positive in their action.

This reworking method has reportedly been used successfully on several transmissions in Pontiac's equipped with factory high-performance engine equipment and used for drag racing. To date, transmission performance has been entirely satisfactory. No failures have been experienced.

B & M AUTOMOTIVE

Bob and Don Spar and Mort Schuman, the brains at B&M Automotive in Van Nuys, Calif., the home of the fabulous Hydro-Stick reworked Hydra-Matic transmissions, are continually on the alert for new ideas and products that will improve the performance and durability of their Hydro-Stick boxes. Among the most important of the improvements the fellows have made during the past year is a switch to two new types of discs for their Hydro-Stick clutches.

The clutch assemblies in any Hydra-Matic transmission are extremely important because they determine the amount of torque, which is one of the ingredients that combine to form horsepower, the transmission can conduct from the engine to the driveshaft. One of the things that determine a clutch's torque capacity is the strength of the friction material on its lined discs. The clutch cannot transmit more torque than this material is capable of holding without wearing rapidly or being torn apart. Experience at B&M indicates that the standard Cork-Krafelt discs used in stock Hydra-Matic's are usable with engines that deliver enough torque for an honest 175 to 200 horsepower. This low power limitation makes these discs useless in transmissions for modern V8's that have been reworked to any extent. They are not used in any B&M transmissions.

One of B&M's special discs has a high-porosity, non-metallic facing that is much more durable than the facing on standard plates. It has been found to be satisfactory for engines that deliver up to 400 horsepower. The facing is grooved in a waffle pattern to increase the amount of oil that can flow across it and carry heat away from the clutch assembly. Keeping a clutch as cool as possible lengthens the life of its parts and increases its power transmitting capacity. A word of warning about these discs: Although they have the same appearance as discs used in Buick transmissions, the composition of their facing material in no way resembles that of the material on Buick discs.

In their Hydro-Stick's for blown engines B&M uses discs that have a semi-metallic facing material. These have proved themselves capable of transmitting 750 horsepower without failing. Facings on these discs are also grooved for oil flow; however, the grooves are not arranged in a waffle pattern as they are on the non-metallic facings.

B&M's special clutch discs are not available individually. The only way they can be bought is as a part of a Hydro-Stick transmission.

An improvement for transmissions installed in Chevys is a special tailshaft that accepts a stock Chevrolet universal joint. Before this shaft was available it was necessary to adapt an Olds joint to the Chevy driveshaft. Being able to use a Chevy joint simplifies the driveshaft reworking.

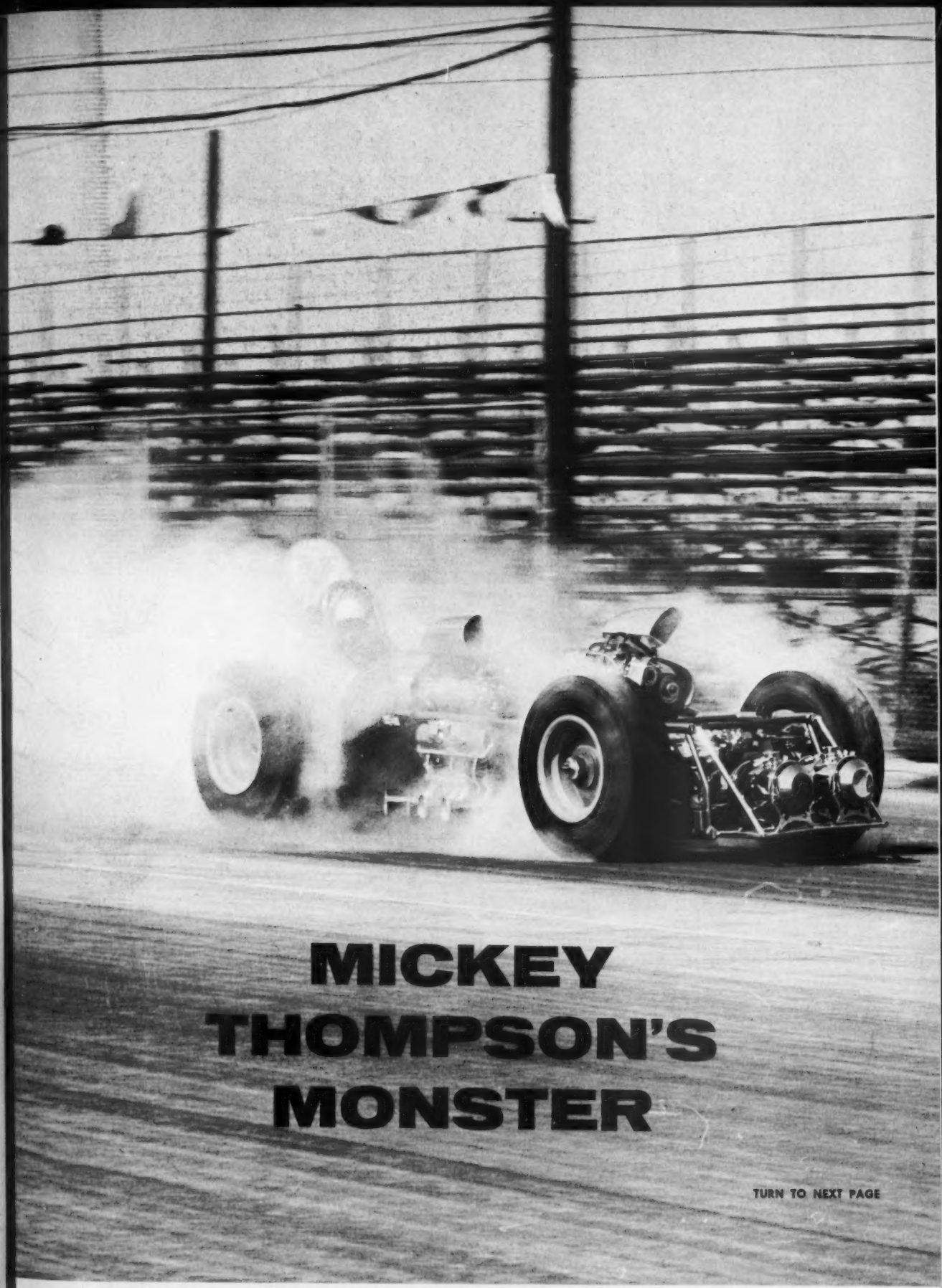
B&M have expanded their line of adaptor equipment that enables a Hydro-Stick or a standard Hydra-Matic to be installed on engines of different makes. Included in the list now are 1953 through '61 Buicks, '49 through '61 Cadillacs, '49 through '61 Oldsmobiles, '55 through '60 Pontiac V8's, '55 through '61 Chevrolet 283 and 348-inch V8's, '51 through '61 Chrysler, Dodge, Plymouth, and DeSoto V8's, four-cylinder Tempests, and '58 through '61 Ford, Mercury, and Lincoln engines of 332 or larger cubic inch displacement. Kits for '61 Pontiacs will soon be available. Because of the rear-mounted transmissions in Tempests, B&M adaptor equipment can be installed only on four-cylinder Tempest engines used in installations where the transmission can be bolted directly to the engine.

B&M transmission adaptor kits consist of all parts necessary to connect the transmission to the engine. Parts included in them vary for different installations. They may have a flywheel, a pilot bearing adaptor, and other parts but one part they all have is an adaptor housing that bolts to the engine's cylinder block. All the adaptor housings have one thing in common. This is that they are designed to accept an Olds Hydra-Matic bellhousing. As this bellhousing will bolt to any of the transmission cases suitable for installations of this type, making the installation in this manner enables any of the transmissions to be used for all but one make of engine. The exception involves '57 through '61 Buicks. Because of the small dimensions of Buick cylinder blocks, a Chevrolet pickup truck Hydra-Matic flywheel, torus cover, and torus assembly must be used with them. As these Chevy parts require a late Hydra-Matic mainshaft and carrier assembly, only transmissions that have these parts can be used for the installation.

Another improvement B&M is developing and that should be available shortly after this book is published is a special oil cooler for any Hydra-Matic installation. Extensive tests with one of their Hydro-Stick's in a car fitted with several pressure gauges and temperature recording instruments showed that a Hydra-Matic's efficiency drops off rapidly as the temperature of its oil goes up. As the oil becomes hotter its viscosity drops; in other words, the oil becomes thinner. This causes the pressure in the transmission's many circuits to drop and this, in turn, reduces the transmission's torque capacity. As the capacity drops, the transmission's friction parts slip a little more, this heats the oil more, the capacity drops even more, and so on.

The only cure for the hot-oil problem is to provide some means of cooling the oil so its viscosity will remain somewhere near what it should be. This will reduce slippage and the vicious circle that causes the oil to become hotter and hotter until the transmission fails. Present plans for the cooler include a small radiator that will be mounted ahead of the engine at some point where it will be exposed to cool air and suitable hoses to connect it to the transmission. Pressure created by the transmission's standard parts will circulate the oil through the cooler.

Reducing or eliminating slippage in a transmission that results from high oil temperature increases the life of the transmission's friction members. This makes one of B&M's oil coolers as much of a necessity on a car used for normal driving as on a competition car. **DON FRANCISCO**



MICKEY THOMPSON'S MONSTER

TURN TO NEXT PAGE

BY THE time someone gets around to writing an article about a competition car, the car has usually collected a name of some sort. So it was with Mickey Thompson's new twin-engine four-wheel-drive dragster. Because of its massive, strong appearance, it's called "The Monster."

Mick built The Monster for attempts on the World's Standing Start kilometer and mile acceleration records in the Unlimited and A classes. He holds the present records in these classes with another car but if someone should break them he wants to be ready to take them back. According to Mick's computations the car that breaks the present records will have to be capable of coming out of the kilo at 251 mph and out of the mile at 283 mph. This is really whistling, and the main reason The Monster is so sturdily constructed. It will also be used for quarter-mile drags.

The Monster was completed in March but so far mechanical failures in its drive assemblies have prevented it from proving itself in test runs on the LADS drag strip in Long Beach, Calif., that Mick manages. Just about every time he puts the spurs to its 1500 horses something happens in one or both of its ring and pinion gear assemblies. Usually the differential gears break or gall in the ring gear carrier. This isn't conducive to driver confidence. But it won't be long before this problem will be eliminated. The boys are going to replace The Monster's two Halibrand quick-change centersections that use Ford straddlemount gears with a pair of centersections, also Halibrand, made for Championship-class race cars and huskier gears.

Mick has a big crew of active rodders in his shop that converts his ideas into roaring iron. One of these fellows is Ken Droesbeck. Ken has been what we could call the project engineer for The Monster. Most of the other crew members have had their chance to contribute to the car but Ken, Roger Beck, and Glen Olsen built most of it.

The Monster is the second twin-engine dragster from this garage but it's the first one to have its engines side by side. Its left engine drives the rear wheels and its right engine drives the fronts. This was accomplished by mounting the left engine in the normal manner but on the left side of the frame's centerline and offsetting the rear axle's ring and pinion gear assembly to align with its crankshaft, and reversing the right engine in the right side of the frame and offsetting the front axle's ring gear assembly to the right to align with its crankshaft. This enabled the crankshafts to be spaced the desired distance apart. With the right engine reversed in the frame, its crankshaft rotates in the direction opposite that of the left engine's shaft.

As Mickey wasn't entranced with the idea of the engines'

So articulated it looks alive, Mickey Thompson's new four-wheel-drive kilometer machine is a masterpiece.

crankshafts being free to rotate independently and drive the front and rear wheels at different speeds, he connected them with what has become the approved method for twin-engined cars by meshing the starter gears on their flywheels. This was simplified by the fact that the shafts rotate in opposite directions but was complicated by the flywheel end of one shaft's being opposite the other shaft's front end. The obvious solution was to fit a flywheel and gear assembly to the front end of one of the shafts to mesh with the flywheel on the rear of the other shaft. Mick went a step farther by fitting a wheel to the front end of both shafts. This enabled both ends of the shafts to be connected and reduced the load each set of gears must carry.

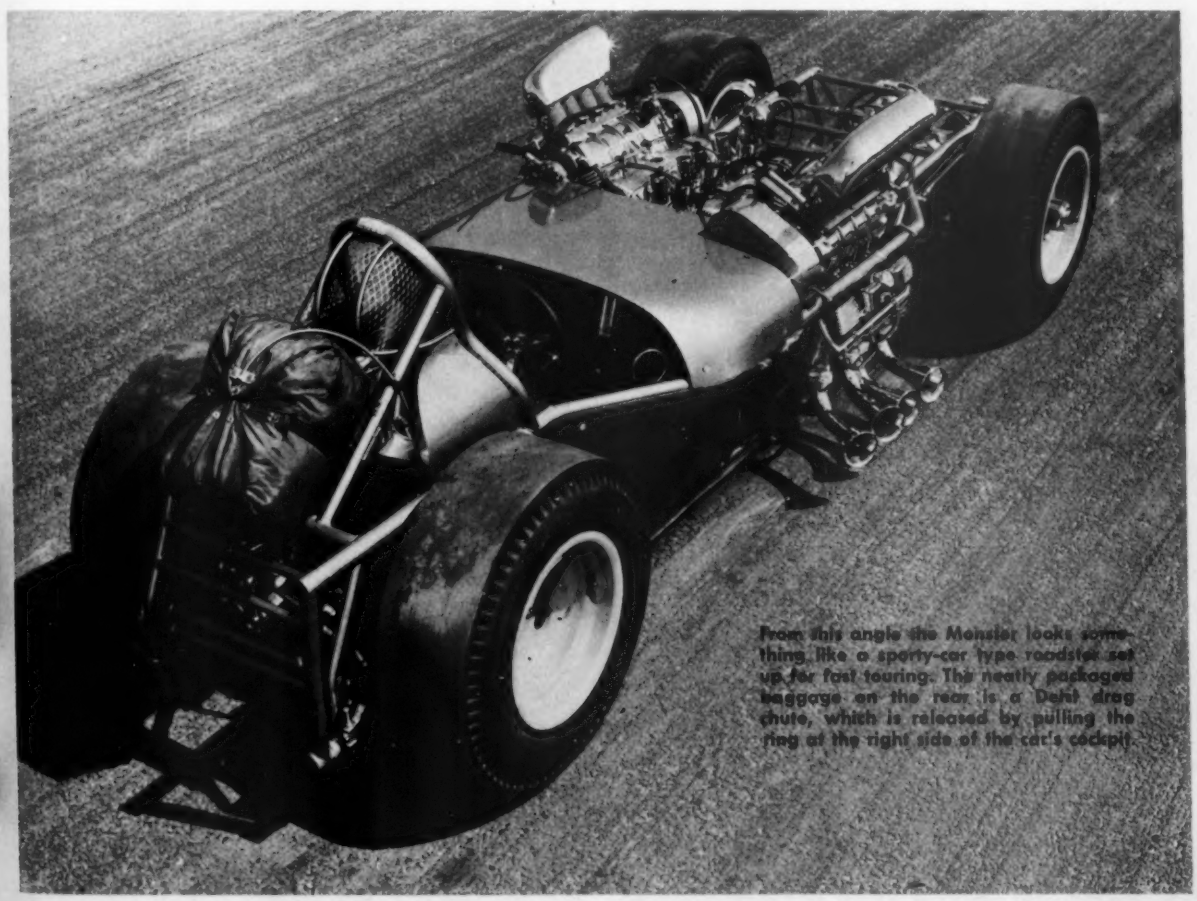
Ken spaced the engines on the ¼-inch thick aluminum plates that support and hold them in their correct relationship to each other so there is a minimum clearance of .020-inch between the ends of the teeth and the bottoms of the grooves in the meshed flywheel gears. He estimated this much clearance would be necessary to prevent the teeth from bottoming when engine and clutch heat caused the Schiefer flywheels to expand. To reduce the loads on the individual teeth, two gears were installed on each of the flywheels. To date the gears haven't given any trouble.

The Monster's space frame, fabricated from SAE 1015 seamless steel tubing, was designed for a wheelbase of 108 inches. Tubing for its lower main members has an outside diameter of 2½ inches and a wall thickness of .089-inch. The upper main members have an o.d. of 1¾ inches and an .089-inch wall. Diagonal and vertical members that connect the main members have an o.d. of 1 inch and a .0625-inch wall. The roll bar complies with NHRA drag rules by having main members of 1¾-inch, .125-inch wall tubing. This tubing is also SAE 1015 seamless steel.

An addition to the roll bar that seems to make good sense is a cage fabricated from expanded metal and small-diameter tubing that surrounds the driver's helmet when he is strapped in the car's upholstered seat with a Deist lap belt and shoulder harness. If the car should flip or roll it would be practically impossible for his head to move to any position where it wouldn't be protected by the roll bar. The seat is just ahead of the rear axle assembly, rather than behind the axle in the standard dragster location.

The shortened Ford housing and axle shafts for The Monster's rear axle assembly, combined with the 16-inch Halibrand magnesium wheels that have 10-inch wide rims, give the car a rear tread width of 36⅝ inches. To offset the gear housing the correct amount to align it with the engine, the left housing and axle were reduced to little more than stubs. Standard Ford hubs were used on the axles but after one of them split on a trial run it was decided to replace them with stronger parts. This will be done at the same time the centersections are replaced. Each rear wheel has a Halibrand dual-spot disc brake.

A Chevrolet torque tube approximately 37 inches long that has a standard slip joint on its front end extends forward from the axle assembly's centersection. The slip joint is secured to a Chevrolet transmission rear bearing retainer bolted to a ⅜-inch thick round magnesium plate that forms a cover for the deep housing that encloses the flywheel and clutch asesmbly. Centered in the plate is a ball bearing that supports a stub shaft on which is mounted the two 11-inch Velvetouch driven discs for the Schiefer



From this angle the Monster looks something like a sporty-car type roadster set up for fast touring. The neatly packaged baggage on the rear is a Deist drag chute, which is released by pulling the ring at the right side of the car's cockpit.

dual-drive clutch. Splines on the stub shaft's rear end match those in Ford universal joints.

Inside the torque tube is a driveshaft with Ford splines on its ends. The spline on the shaft's axle end fits into a standard Ford driveshaft coupling sleeve on the axle assembly's input shaft. The spline on the engine end supports a Lincoln needle-bearing universal joint that has been fitted with safety straps to prevent its being thrown apart if it should break.

The clutch's pressure plate assembly is a standard Schiefer with an aluminum plate. It is a semi-centrifugal type and its springs exert a total of 2800 pounds pressure. A steel floater plate separates the two driven discs. The flywheel and clutch housing, acting as a scattershield, was made from ¼-inch thick steel plate. It also holds a shaft for the clutch's release mechanism.

Ford and Willys Jeep components were ued for the front axle assembly. The assembly's centersection was offset to the right about 7½ inches. Its longer left axle housing was made from 4 inch o.d. steel tubing. With the same type wheels as those on the rear, it provides a tread width of 49½ inches. Its driveshaft and torque tube assembly is identical to the one for the rear axle with the exception that the torque tube is only 15½ inches long. The engine's flywheel housing and clutch and flywheel assembly are identical to those for the left engine. Front wheel alignment specifications are 12 degrees caster, 0 degrees camber, and 3/32-inch toe out. Toe out rather than toe in is required because the wheels are pulling rather than being pushed.

Ring and pinion gears in both the front and rear centersections have a ratio of 3.78 to 1. The final ratio used for most test runs made so far has been 3.48 to 1. An adjustable bronze thrust button in the axle housing on the ring gear side of each centersection provides support for the ring gears when the pinion gears are transmitting heavy torque loads to them. These buttons are positioned so that they are behind the gears, directly opposite the pinion gears.

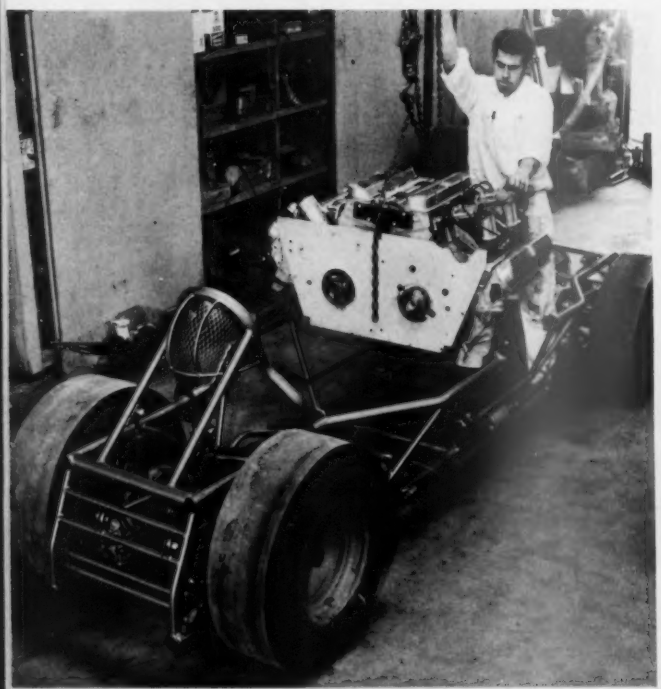
Fitted to the outer ends of the axle housings are Jeep steering joints that support the wheels and enclose the universal joints and stub shafts that drive the wheels. Secured to the top of each joint by six capscrews is a long steering arm, cut from 4130 chrome-moly steel plate, that extends forward to a short tubular tie rod linked to the rack in a Jaguar rack and pinion steering gear. The steering gear is mounted on a bracket welded to the front side of the axle assembly's left housing. The tie rods were made from ⅞-inch o.d., .0625-inch wall 4130 tubing. Threaded sleeves for ⅝-inch Heim joints were welded into their outer ends. A long steering shaft of 15/16-inch o.d. 1015 tubing extends back from the steering gear to a 14-inch diameter rubber-covered steering wheel. The shaft's only support is a self-aligning ball bearing bolted to the rear engine support plate.

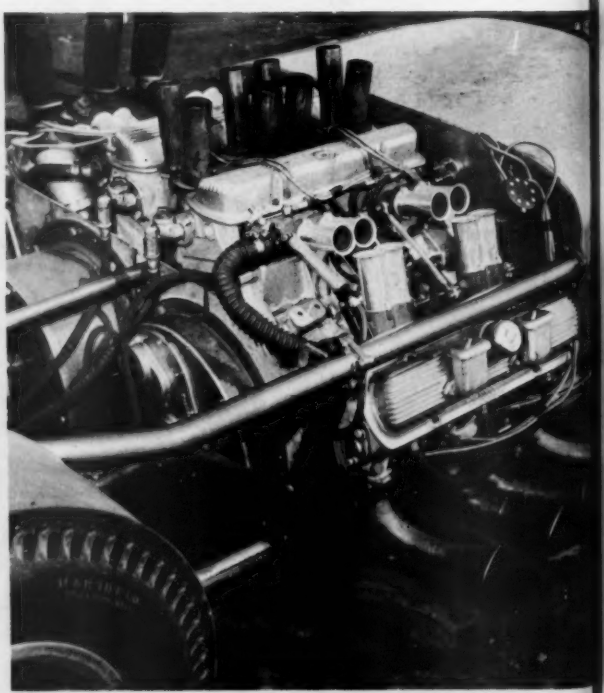
The Monster differs from most dragsters by having full torsion bar suspension systems for both its front and rear axles. The torsion bars were made by Halibrand. Those for the front axles are enclosed in tubular crossmembers welded to the front end of the frame below and ahead of the axle. Arms cut from 4130 flat stock and approximately 15 inches long extend back from the bars to brackets welded to the underside of the axle housings. Short arms fitted with adjusting bolts enable the anchored ends of the bars to be rotated to raise or lower the frame in relation to the axle.

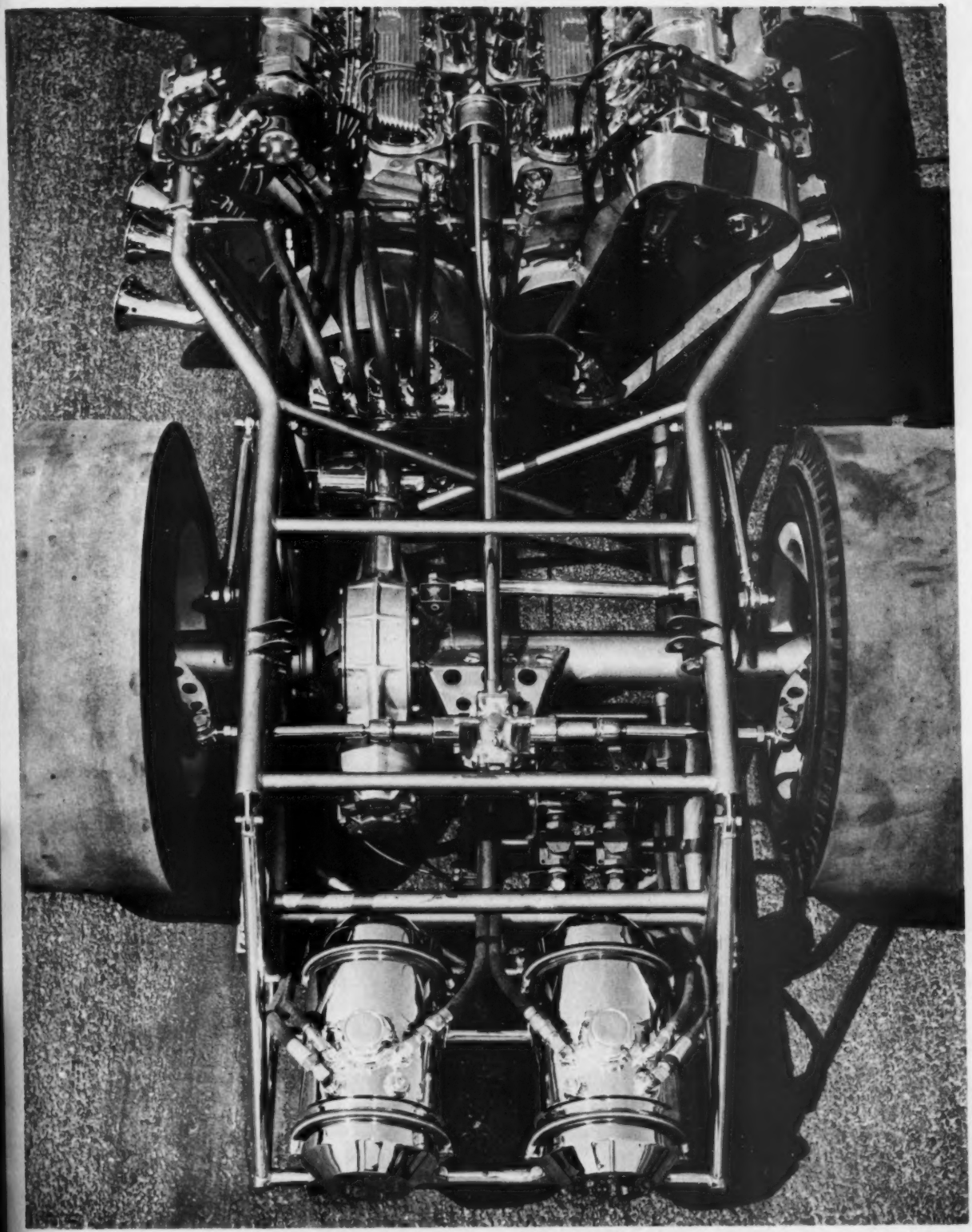
To hold the axle assembly in its correct fore and aft location and prevent its rotating when the assembly's pinion gear is pushing hard on the ring gear, two control rods approximately 15 inches long, fabricated from ⅞-inch o.d., .0625-inch wall 4130 tubing and ⅝-inch Heim joints, extend back from brackets welded to the top of the axle housings to brackets on the frame.

Because a control rod and torsion bar arm arrangement of the type used on The Monster provides rigid control of the axle assembly's position in the frame under all conditions, the geometry of the rods and arms must be correct when the arrangement is used with a torque tube drive so that the input shaft for the drive gear's pinion shaft will be aimed directly at the center of the driveshaft's universal

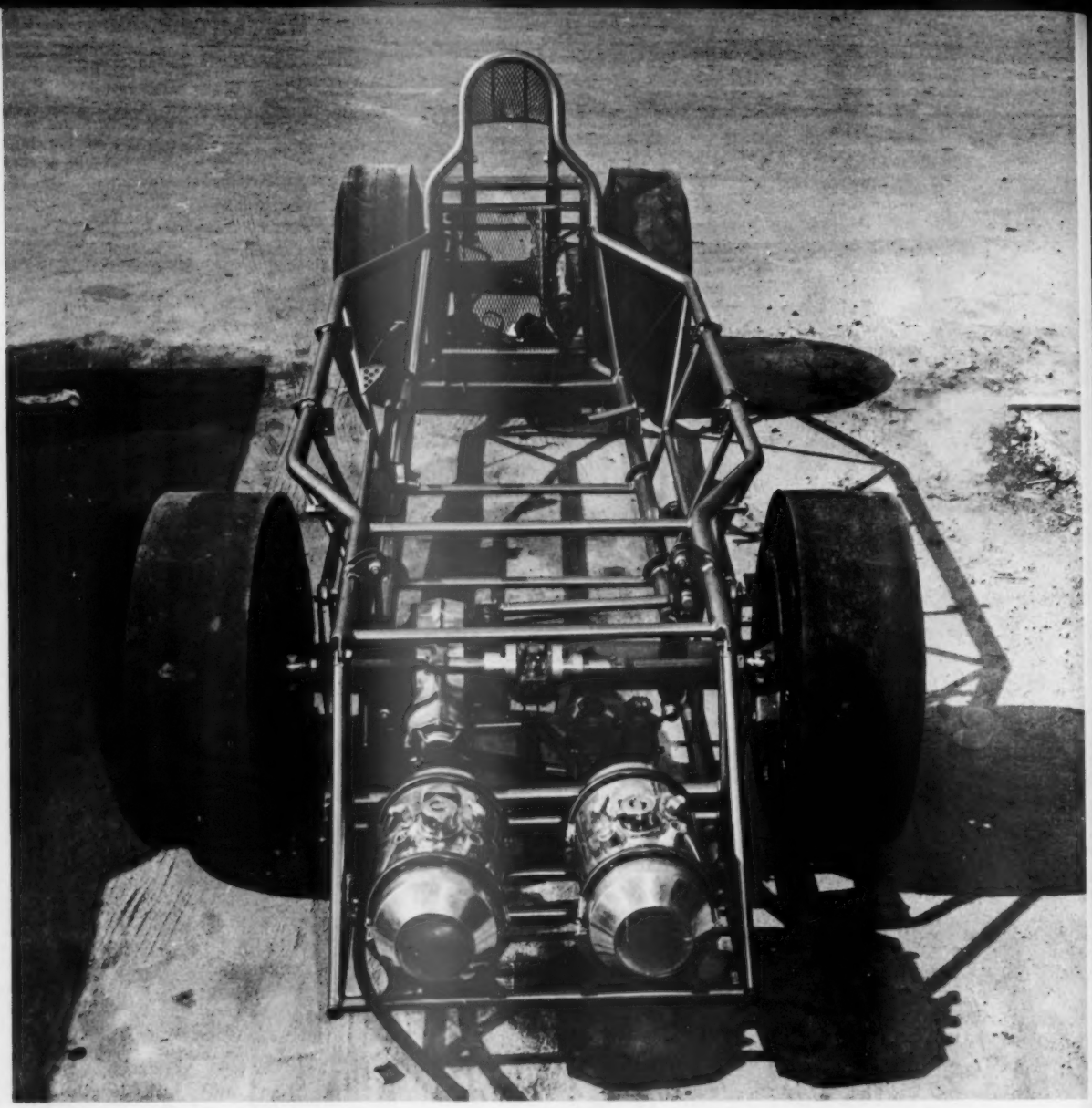
Photo below left shows side-by-side engines being lowered into place. Close-up at right gives fine view of the twin 1960 Pontiac V8's. They have been reworked to the hilt and fitted with 6-71 GMC blowers. Left engine drives rear wheels; right engine, fronts.







Heavy use of chrome adds definite eye appeal to what is a functional type machine.



joint at all times when the frame moves up and down in relation to the axle. Any attempt of the arms to twist the axle assembly out of this alignment as the frame moves would be resisted by the torque tube and driveshaft. This would place unnatural strains on the chassis and axle parts that could become severe enough to damage the parts. The simplest method of eliminating the possibility of such misalignment would be to use an open driveshaft that has a universal joint at each of its ends. The only purpose of a torque tube that has a slip joint is to prevent gear torque from rotating the axle housing. With this suspension setup, rotation is resisted by the suspension members.

Torsion bar action is controlled by a pair of Monroe double-acting tubular shock absorbers mounted just inside the main frame members and in approximately vertical positions. Lower ends of the shocks pivot on studs welded to the underside of the axle housings; their upper ends are secured to brackets welded to upper main frame members.

Lateral control of the axle assembly is accomplished by a rod similar to those for fore and aft positioning, except it is longer, that has one end secured to a bracket on the left side of the frame and its other end secured to a bracket welded to the gear housing flange on the left axle housing. This rod is behind and parallel to the axle housing. It resists any tendency of the axle or frame to move from one side to the other.

The suspension system for The Monster's rear axle assembly has components similar to those for the front axle but the components are arranged a little differently. The torsion bars are mounted behind and below the axle and the frame ends of the control rods are also behind the axle. Axle ends of the rods connect to brackets welded to the upper side of the right axle housing. With this arrangement the torsion bar arms and the control arms form parallelograms. The shock absorbers are vertical and quite close together. Lateral frame and axle movement is resisted by a rod with one end secured to a bracket bolted to the torque tube flange on the gear housing and its other end attached to a bracket welded to the lower right frame member.

Tires that support The Monster's four corners are identical. They are 9.00 x 16 M&H Racemaster Dragsters. Their

tread surfaces are nine inches wide and they are inflated to about 35 pounds, depending on the strip.

Twisting effort that makes its tires look like four smudge pots when Mick stands on The Monster's throttle comes from two 1960 Pontiac V8's that have been reworked to the hilt and fitted with 6-71 GMC blowers.

Originally the engines had displacements of 389 cubic inches but replacing their crankshafts with C-T Automotive ¼-inch chromed strokers that have strokes of 4 inches boosted this to 414 inches. Cylinders were left at their standard 4 1/16-inch bore diameter so their walls would have maximum strength to resist the high combustion pressures that would be created in them. The cylinder blocks are the high-performance option that have four-bolt main bearing caps. As these caps are strong enough in their standard form for any reworked engine, it wasn't necessary to replace them with steel caps or strengthen them in any way. Clevite 77 bearing inserts were used to support the crankshaft.

Mickey manufactures forged aluminum connecting rods and pistons for Pontiacs. These were installed in both engines. McQuay-Norris piston pins, which have thicker walls than standard pins, link the pistons to the rods. Compression height of the pistons give the cylinders a compression ratio of 8.5 to 1. Piston skirt to cylinder wall clearance is .011-inch. The rods have Clevite 77 locked-in bearing inserts.

Mickey recently started grinding camshafts for Pontiacs, in both roller tappet and flat tappet types. He will soon have complete cam kits, including a shaft, tappets, and valve springs, for Pontiac owners. Hidden deep in each of The Monster's cylinder blocks is one of his steel-billet roller-tappet shafts and sixteen of his roller tappets. GMC tubular pushrods transmit tappet movement to stock Pontiac 1.6 to 1 ratio rocker arms mounted on 1960 cylinder heads. The rocker arm studs were pinned in their bosses to prevent their being pulled out of the heads by valve spring and pushrod pressure. Actually, the thing that usually lifts these studs, in either Pontiac or Chevy heads, is valve float. The sharp jolts the rocker arms give the studs when the valves close under float conditions are much more severe than the pressures exerted by the arms normally.

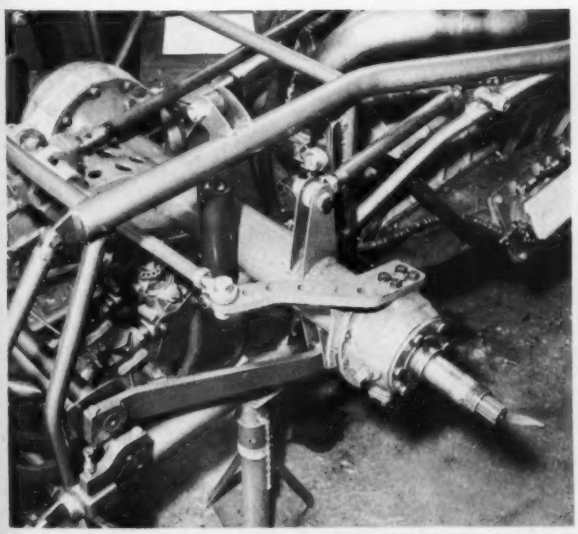
Before he started grinding his own camshafts Mick didn't like to run his V8's over 6500 rpm. This was usually over the engine's peaking speed and getting too close to serious valve float conditions. He says that dynamometer test engines fitted with his shafts have turned 7100 rpm, with good power output and no indication of valve float. As yet the engines haven't been turned any tighter than this but plans are to keep boosting the rpm until the maximum, as limited by valve float or some other condition, is reached.

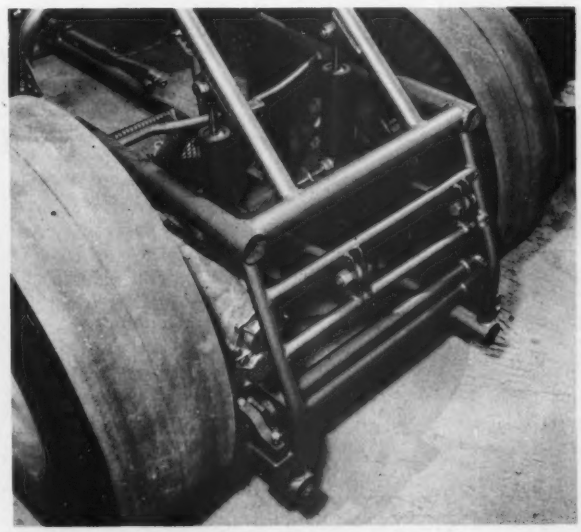
Intake and exhaust ports and passages in the heads were enlarged by Jocko's Porting Service in Long Beach, Calif., to allow the engines to take full advantage of their blowers. Standard intake and exhaust valves were replaced with special lightened and polished 1/16-inch oversize valves. Valve face angles are 30 degrees on the intakes and 45 degrees on the exhausts. Mickey's special dual valve springs are locked to the valve stems with heavy-duty retainer washers.

Mounting the engines side by side and at the 30 degree angle placed the heads on their inner cylinder banks so close together that there wasn't space to bolt the exhaust pipes to them. This little difficulty was overcome by machining the standard exhaust flanges off the heads and welding 90-degree cast-iron elbows to the ports. The upper ends of the elbows have flanges to which 3½-inch lengths of 2-inch o.d. tubing are bolted. The exhaust flanges on the outer heads were left as-is. Header adaptor flanges and individual 2-inch o.d., 12-inch long exhaust pipes with belled ends are bolted to them. These flanges and pipes are two more of the builder's products. The purpose of the flanges, which fit between the heads and the pipes, is to simplify the installation of large pipes on the ports by providing a larger bolt pattern than the standard pattern in the heads. The flanges bolt to the heads, pipes bolt to the flanges.

Tilting the engines outward in the frame lowered their outside heads so far that oil in the heads' rocker arm chambers wouldn't flow through the standard drain passages

Details of Monster's space-type frame are shown in the photo at left. All the frame members were made from SAE 1015 seamless steel tubing. Following conventional dragster practice to provide the best handling qualities, the tread of the rear wheels was made narrower than the front tread. Shown in photo below are details of the torsion bar suspension system and the Willys front wheel drive units that were adapted to the car's front axle housings. Monster differs from most dragsters by having full torsion bar rear suspension, below right.





and return to the crankcase. New drain holes, three in each head, were drilled through the low side of the chamber walls and threaded with ¼-inch pipe threads. Drain pipes formed from ½-inch copper tubing were connected to fittings screwed into the heads and oil pan.

To keep the oil in the pan of the reversed right engine within reach of the oil pump when the car is accelerating, the pan's sump was extended to what is now the rear of the pan and fitted with baffles. The pickup tube on the pump was lengthened to place its end close to the rear of the sump. Standard full-flow oil filters, in their stock locations, are used on both engines. Oil is SAE 40 Valvoline.

Noel Timney, who spends his working time at Signal Hill Machine Shop, Signal Hills, Calif., made the chain drive setups for the 6-71 blowers. A two-row roller chain, driven by a sprocket connected to the front end of the crankshaft, rotates the impellers in the blowers at 1.1 times crankshaft speed. Intake manifolds designed and produced by the Thompson group support the blowers. Each blower is said to effect a maximum boost of approximately 17 psi over atmospheric in the engine's induction system.

Mounted on each blower is a Hilborn fuel injector. Each injector has four 3-inch diameter air inlets. For test runs the car has made so far the injectors have been calibrated for both gasoline and methanol-base fuels. The aluminum air scoops fitted to the blowers were made by Don Borth. Don also formed the car's aluminum body panels.

Fuel is carried in twin Eelco 3½-gallon aluminum tanks mounted on a framework fabricated from steel tubing and bolted to the front end of the frame. Behind the left tank are an aircraft shutoff valve, actuated by a control in the cockpit, and an aircraft fuel filter for each engine. Hoses that have an inside diameter of ½-inch connect the Hilborn fuel pump for each engine with its tank and filter and valve assembly.

A Hilborn fuel pump and a Jabsco "Water Puppy" water pump for each engine are mounted on drive adaptors, another product of Mick's, that bolt to the rear ends of the blowers. These adaptors support the pumps, connect their shafts to blower impeller shafts.

Water pumps on automobile engines are usually associated with radiators but The Monster doesn't have even one radiator, not to mention two of the heavy things. All The Monster's pumps do is circulate water in the engines through the cylinder banks and heads to prevent the formation of steam pockets near hot spots in the combustion chambers. This is an excellent idea. Steam pockets could allow the temperature of combustion chamber hot spots to climb high enough to pre-ignite the compressed fuel and air mixture in the cylinders. This could cause trouble.

A small catch tank mounted on the right engine's flywheel cover, just below the driveshaft, is connected to the high cylinder head of each engine with a hose to provide expansion room for the water in the engines when the water's temperature rises. Hilborn injector fuel line check valves in the hoses prevent pressure created in the engines' water jackets by the pumps from forcing water into the tank.

Ignition systems consist of Joe Hunt Scintilla Vertex magnetos, Autolite spark plug cables, and usually Champion J-58R spark plugs. Advance weights in the mags are limited to 26 crankshaft degrees. An initial lead of 12 degrees gives a maximum ignition advance of 38 crankshaft degrees.

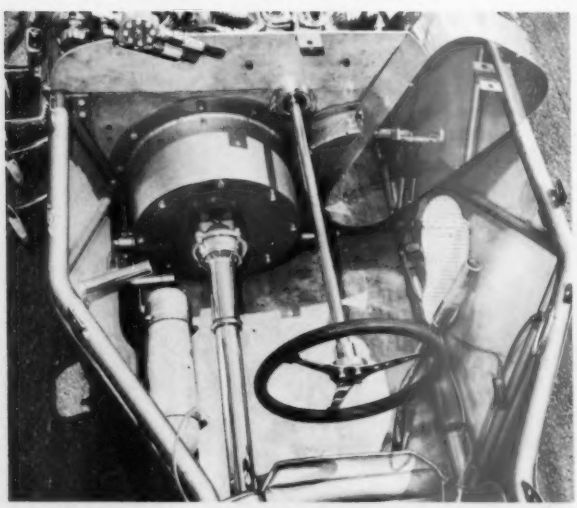
In The Monster's cockpit, in addition to the steering wheel, are a foot-actuated clutch pedal, a hand brake lever that actuates the Micro two-stage master cylinder for the brakes on the rear wheels, a Moon throttle pedal that controls the Marbet hydraulic throttle linkage for the fuel injectors, and release ring for Deist drag chute used when necessary. There's also an oxygen bottle and mask setup that the driver can use when the car is sitting on the starting line. When the car is facing into a headwind exhaust from its inside banks make breathing difficult.

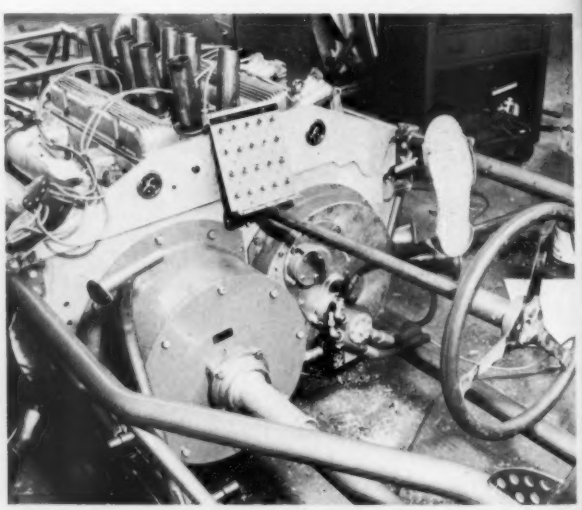
Instrumentation includes an oil pressure gauge for each engine, mounted on the engines but where they can be seen by the driver, and a Jones tachometer. The tach is driven by the front end of the left engine's crankshaft and is mounted on a bracket bolted to the front engine support plate, directly in the driver's line of vision.

The Monster's massive appearance wouldn't lead anyone to believe the car is a flyweight, and it isn't. With Mick's 210 pounds in the seat it weighs 2940 pounds—fuel, oil, and water aboard. The front wheels carry 1610 of these pounds and the rears take 1330.

Because it was built for such a specialized purpose, The Monster may graze for some time before the world learns whether it is capable of performances of World's Record caliber. But in the meantime it will be let out now and then on various drag strips to keep its muscles in shape.

DON FRANCISCO





The Monster's driver sits ahead of the rear axle assembly. The lever near the right side of the steering wheel actuates the two rear brakes. At the left of the cockpit is the long clutch lever that has ample leverage for the Monster's two extra-long clutches.

PROBLEMS?

BATTERY-POWERED ignition systems have been around since the early days of the automobile when they eased the magneto out of the picture by virtue of easier starting plus simpler and cheaper construction. The basic system changed little from the early 'twenties to the early 'fifties but when modern overhead valve V8's came into the automotive picture, some serious problems quite suddenly arose.

The overhead valve V8 was a much more efficient engine than its flathead predecessor but the higher compression ratios and increased engine speeds that produced efficiency also presented new problems for today's ignition systems. Higher compression called for better anti-knock qualities in the fuel but the lead-bearing additives which prevented knocking aggravated the spark plug fouling problem. Wider spark plug gaps to promote smooth idling and increase economy also imposed added strain on the already taxed voltage available to fire the plug.

Voltage required to fire the spark plug went up as compression ratios were increased and plug gaps were widened but with the conventional battery ignition system, available secondary voltage drops as the engine speed increases. So the result was often inadequate voltage to fire the plug at higher engine speeds. Partial relief to the problem of poor high-speed performance was achieved through the use of ballast resistors, increased dwell time and twelve-volt systems. The limiting factor to increased secondary voltage with the conventional system, however, is the current carring capacity of the contacts.

Modern breaker points have tungsten contacts with a melting temperature of 6100° F. When the contacts are closed, current flows through the primary winding of the coil creating a magnetic field. When the contacts open, this magnetic field collapses, inducing a high voltage in the secondary winding of the coil. Each time the contacts open, current density increases to the melting point of tungsten

and creates a molten bridge of metal which is reduced in cross sectional area as the contacts continue to separate. Finally the current density becomes so great that the metal actually explodes, leaving a build-up of metal on the negative contact and a corresponding crater on the positive contact.

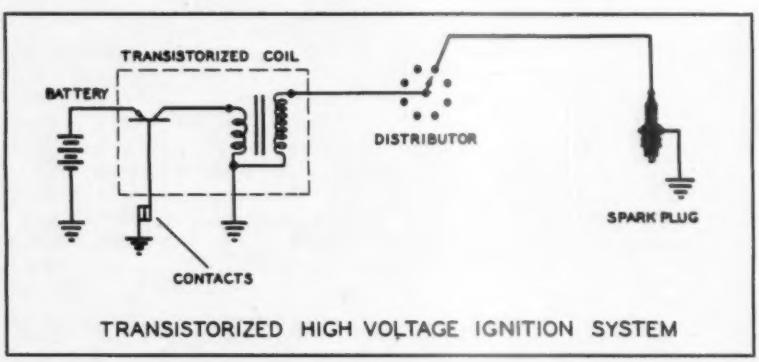
With this action taking place 4 times every revolution on a V8 engine, it is plain to see that contact life expectancy is governed by the amount of current that passes through them. The larger the current the shorter the life. For reasonable contact life, the maximum amperage must be limited to 5 amps. In the pre-high-compression era, 5 amps was more than enough to do the job but with today's engines, it is borderline at high rpm's.

Another problem with conventional ignition systems which has caused many a temper blowup on cold mornings is what the ignition engineers call "blued points" or "blued contacts." This happens when cold weather causes the engine to turn over so slowly that the contacts stay closed from 40 to 400 times as long as at normal running speeds. The longer contact time plus less resistance in the ballast and cold copper of the coil raises a large area around the point of final contact to the red heat (2000°F) of tungsten, causing blue oxide to form on the breaker points. Oxide is a poor conductor of electricity so current carrying capacity of the contacts is severely restricted, causing hard starting and limiting engine performance.

Since the current-carrying capacity of tungsten contacts reached its practical limit at 5 amps, higher compression ratios and more efficient engines were stymied until something revolutionary could be developed in the field of battery ignitions. The Research Department at the Electric Autolite Company in Toledo, Ohio, decided several years ago that a new means of switching the primary current would have to be developed to overcome the limitations imposed by the contacts.

Electron tubes were considered to switch the current but presented many design problems. They are fragile, require filament warm-up time, require high plate voltage which must be closely regulated, have poor output at low battery voltage (cold starting), would require complex circuitry and the cost for such a system would be high.

Transistors offered a possible solution to many of the objections presented by the electron tube but also presented some disadvantages of their own. Advantages offered by the power transistor included: Ruggedness (no glass envelopes); no warm-up time required; operation at low voltage (6-, 12-, or 24 volts); operation over a wide range of voltage from cold starting to normal operation; and miniaturized circuitry could be used. Disadvantages included:

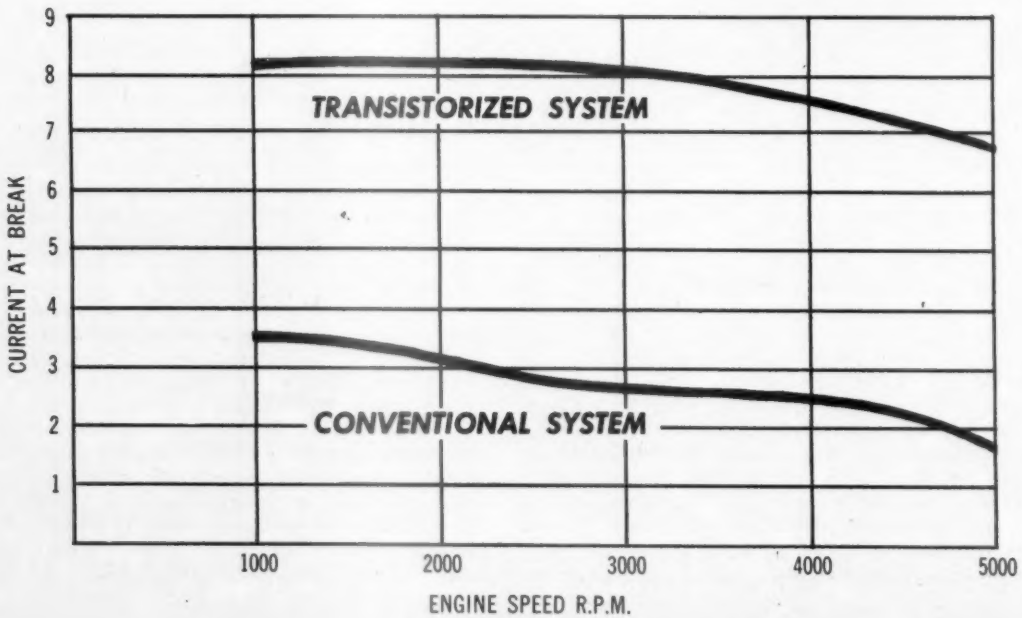


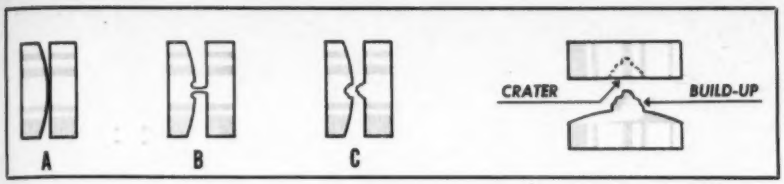
TRANSISTORIZED HIGH VOLTAGE IGNITION SYSTEM

Left—Simplified diagram of Autolite's Transigniter circuit shows how conventional contacts trigger the transistor.

Right—The oil-filled Transigniter coil and attached ballast mount in engine compartment, heat sink in cooler spot.

Below—The two curves show the amount of current available in the primary circuit with conventional and transistor systems.





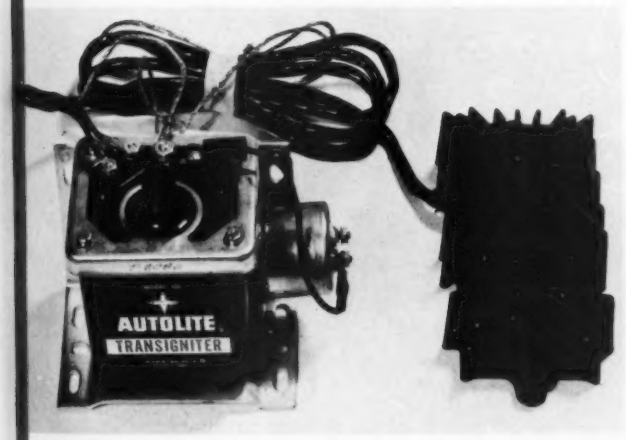
High current through tungsten contacts causes metal bridge to form at break (B) then melt, causing metal transfer.

More complex circuitry than conventional battery systems; higher cost; and with present transistors, they could be used with negative ground electrical systems only.

Two types of power transistors were available for Autolite engineers to try; silicon and germanium. The silicon transistors had temperature and voltage characteristics to handle the job but were ruled out for commercial application due to their high cost.

The germanium power transitor was enough cheaper to offer commercial possibilities and was rated as high as 30 amperes but lacked the ability to carry voltages generated in the primary circuit of a conventional ignition system and would not tolerate the high underhood temperatures often reached in an automobile.

Early in 1956, Autolite decided to proceed with development of a new high-energy ignition but to stay within the limitations imposed by then available germanium power transitors. It was felt that if silicon transitors became more economically attractive at a later time, basic circuits and ideas could be readily adapted to substitute them for the germanium transistors.



All design problems for use of the germanium power transistor could be met in a relatively straightforward manner except for destruction of the transistor due to high voltage in the primary circuit. With conventional induction-type ignition coils, back voltage in the primary circuit reaches 200-300 volts, enough to destroy germanium transistors. Reduction of back voltage on the induction coil was impossible so it was necessary to consider a high tension transformer or "slow coil" which did not have high transistor-damaging primary voltages. The "slow coil" was not a new idea; it had been used back in the early 'twenties but was abandoned in favor of the smaller, lighter and cheaper induction coil. The induction coil also had a 1500 or 2500 cycle secondary output which had showed a much better ability to fire carbon fouled spark plugs than the 500 cycle secondary of the "slow coil."

A return to the "slow coil" with low back voltage in the primary would permit use of the germanium transistor and increase contact life but according to past theories, with its lower secondary cycle it would give less ability to fire fouled spark plugs. Autolite then launched a test program to determine the ability of both the induction and the "slow coil" systems to fire fouled plugs. Contrary to the theory established in the early 'twenties that the induction coil would fire fouled plugs better than the "slow coil," the two types were found to be equal in ability and both would fail to fire after the same interval of time under an accelerated "fouling" program. Variations in combustion chamber temperature, lead content of the fuel, spark plug heat range and fuel distribution controlled the amount of "fouling" deposits on the spark plugs and determined whether a system would or would not fire the plug.

Assured that the "slow coil" could hold its own with the induction coil in ability to fire fouled plugs, further development using the germanium power transistor prceeeded. Transistors have the remarkable ability to amplify current greatly and since it is a solid composition, will not wear or deteriorate like mechanical switching devices.

Since the limiting factor of conventional ignitions had been the ability of the tungsten contacts to pass more than 5 amps current, the transistor was used to replace the contacts in switching the heavy current but the tungsten contacts were retained in the system to trigger the transistor. A very low .25 amperes could be passed through the contacts to trigger a 7.5-amp load from the emitter to the collector of the transistor. Thus, in the transistorized system the tungsten contacts carry only 1/20 of the current they handled in a conventional system and no longer restrict the ignition's ability. With such a small current through the contacts, a condenser to prevent arcing is no longer needed. Contact life is no longer governed by erosion but rather by the rubbing block that opens the contacts. An occasional adjustment is needed to compensate for rubbing block wear but with proper lubrication it should last well past the 50,000 mile mark.

The secondary output of the transitorized system is relatively flat throughout the operating range of the engine because of a fast rise in current due to the relatively low primary inductance in the new system. Whereas the conventional battery ignition system has approximately 50% less available voltage at 5000 rpm than at idle, the Autolite's transistorized ignition varies less than 5% over the same range. This ability insures adequate voltage at all engine speeds regardless of modern compression ratios or spark plug gaps.

Should more secondary voltage be required, slight changes in design can be easily made to raise the voltage much higher while still retaining the advantages of a flat output curve. Present limitations in insulation between contacts in the distributor cap and rotor as well as conventional secondary wiring used on our modern automobiles prevent the secondary output being raised higher. Too much voltage with the present insulating practices would result in "flashover" within the distributor or between cables which could damage both ignition and engine.

The transistorized ignition developed by Autolite and soon to be available for the public is called Transigniter and is the final refined version of several prototypes that have been tested for the past four years. It has an oil-filled coil package which is fastened under the hood near the

distributor and a separate finned aluminum transistor and diode heat sink which must be mounted in a location that will not exceed 180° F. The coil unit and attached ballast resistor can easily stand temperatures up to 350° but the transistor is sensitive to heat so a six-foot cable on the heat sink permits mounting in a cooler place than in the engine compartment. Ideal locations for the heat sink are ahead of the radiator baffle behind the grille or on the passenger side of the firewall. The heat sink does not get hot and is absolutely silent so will not disturb passengers within the car.

Immediate applications for the Transigniter include stationary engines which often run for long periods without maintenance, commercial fleet truck operations where "down" time for ignition repairs is costly and for the performance minded automobile owner. Future plans call for the transistorized system to fit into a program for more promising "anti-fouling" ignition systems.

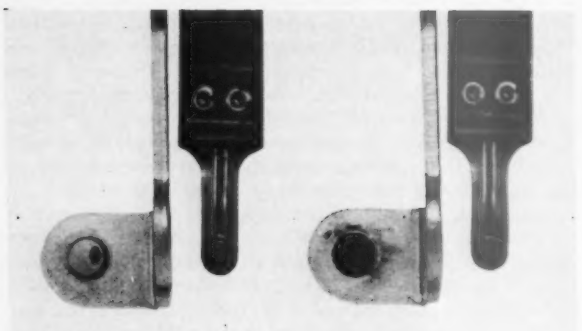
For use with one of today's high-performance cars, the Transigniter is very well suited; no contact wear and flat output curve so no misfire due to lack of voltage are the most noticeable advantages but the price also promises to be good. We don't have a firm price yet but it should be in the neighborhood of $75, which is competitive with dual-coil ignitions and cheaper than a magneto.

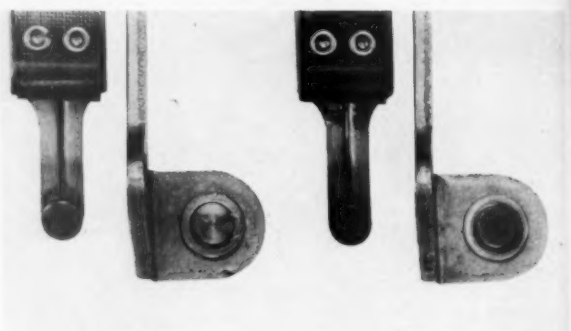
The Transigniter will be available only in 12-volt negative ground versions initially. Negative ground only is dictated by the polarity of the transistor but since most of the late American automobiles use a negative ground electrical system, this is no serious problem. To hook up the Transigniter on a late model car, simply replace the conventional coil, ballast resistor and condenser. Late model Ford and General Motors cars use a resistor type primary lead wire between the ignition switch and the coil instead of a ballast resistor and this must be bypassed to get a full twelve volts when using the Transigniter since it has its own resistor.

For best results at high speeds, Autolite suggests replacement of the standard breaker points with their new Power Point breakers. Distributors with dual breakers should have one set removed; they are not needed for increased coil saturation with the transistor system. The Transigniter oil-filled coil is much heavier than a conventional induction coil so care must be taken to insure it is securely mounted. As mentioned previously, the transistor heat sink must be mounted in a cool place and the 6-foot leads connected to the coil. That's all. Point cam angle dwell and advance are set the same as always.

That magical little item, the transistor, has been responsible for many changes in recent years, and for those of us interested in automobiles, this might be one of the most important. The door is apparently open for even more improvements in modern internal combustion engines, and it might not be too long before we see much higher compression ratios and more efficient engines. That's what we are waiting for.

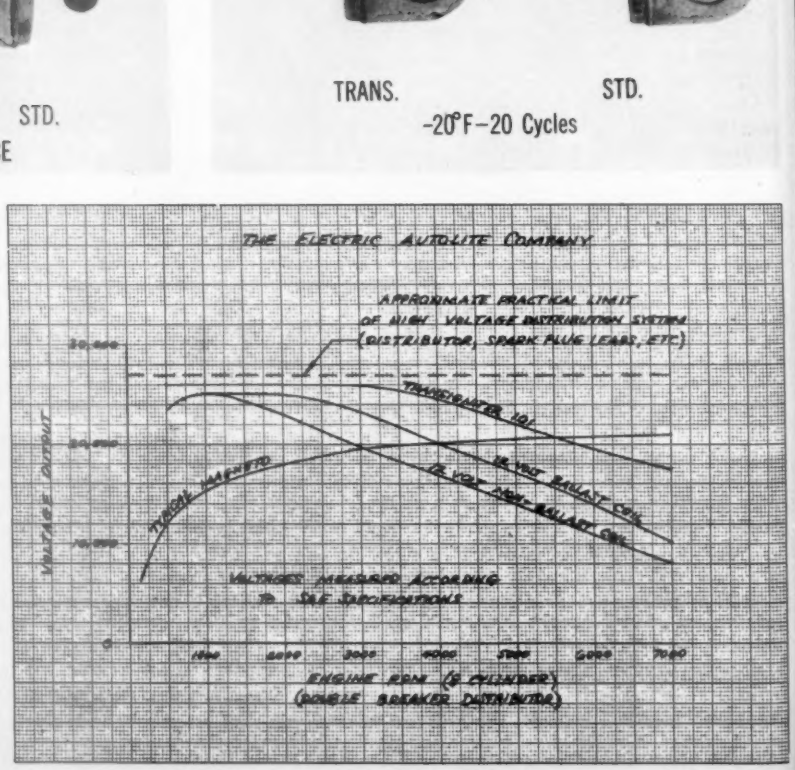
RAY BROCK





Above—Two sets of breaker points after 44,000 miles. The contacts used with the Transigniter, left, are still in good shape, but the contacts used with standard induction coil are deeply pitted, worn out.

Above right—Blue tungsten oxide was formed on contacts used with a conventional ignition, right, after twenty starts at minus 20 degrees F. These contacts are ruined. Contacts used with Transigniter are still in good shape after same testing.


THE ELECTRIC AUTOLITE COMPANY
APPROXIMATE PRACTICAL LIMIT OF HIGH VOLTAGE DISTRIBUTION SYSTEM (DISTRIBUTOR, SPARK PLUG LEADS, ETC.)
TRANSIGNITER 101
12 VOLT BALLAST COIL
12 VOLT NON-BALLAST COIL
TYPICAL MAGNETO
VOLTAGES MEASURED ACCORDING TO SAE SPECIFICATIONS
VOLTAGE OUTPUT
ENGINE RPM (8 CYLINDER)
(DOUBLE BREAKER DISTRIBUTOR)

Right—Available secondary voltage for four types of ignitions. The conventional dual-point ignition with induction coils drops off badly at high rpm's. Magneto has good high rpm output, low starting voltage. Transigniter has high initial output, drops off slowly above 3000 rpm, but stays above maximum required voltage.

FLOATERS FOR FAST CARS

A FEW years ago, a national stock car champion told us that the only stock items on the chassis of his car were the master cylinder and the steering gear. This was not an admission that he was a cheater but it was to illustrate the fact that stock car chassis must be modified considerably to withstand the severe abuse they are subjected to on race tracks. All organizations that sanction stock car races permit and even demand that various components be beefed up to many times their stock strength.

This need for heavy-duty pieces related to stock car racing has been responsible for many shops to specialize in designing and building heavy-duty components for several types of cars. One such shop is Holman and Moody of Charlotte, North Carolina. We stopped in to see the H & M operation last winter on the way home from Daytona's Speedweeks and were amazed at the number of items being made just for racing.

One item that caught our eye was a full-floating rear axle housing for late model cars. For those of you who do not fully understand just what a full-floating ("floater" in racing jargon) axle is, here is a simple explanation. All of our conventional U.S. automobiles use a steel axle housing with the outboard end holding a bearing which supports the axle shaft. This axle, in turn, must support the weight of the car at that particular point. This is a semi-floating axle. A full-floating axle carries only the driving force to the wheel while the weight of the car is supported by bearings on the housing. The obvious advantage of a floater is the fact that the axle shaft only has to drive the wheel. Shock loads of bumps and chuck holes are not absorbed by the axle shaft; the sturdier housing takes the punishment.

An added advantage to a racing type floater is the much larger bearings which can be used. Stock rear axle bearings are designed for cruising down the highway and generally fail early when subjected to the speeds, side thrust and shock loads encountered on a race track. Full-floating axles have large double-row bearings which are better suited for the extremes of racing.

Although the floaters we saw at H & M were designed for stock car racing, there are some definite advantages that should make them appealing to gas classes, modifieds and even dragsters at the strips throughout the country. First of all, the floaters have a true safety hub. With or without axle, the wheel supports the car. Secondly, it is a simple and quick operation to pull both axles from the housing so that the car can be towed on the ground without ring and pinion and driveline turning. Towing a heavy car on the ground with a good tow bar is much safer than using a trailer. Thirdly, there is less rolling friction with the large floater bearings than with stock bearings.

When preparing these units for stock cars, H & M cut off the stock backing plate flange and bearing bore, then replace it with a special floater stub which is welded to the axle housing. At this point, note that the rear wheel tread could also be narrowed if this were for a special drag machine. The stub is machined from chrome moly steel and one model is made to accept the stock backing plate. Another model requires reworking the backing plate. The stub has an inner bore slightly larger than the axle shaft and the outer bore is machined to accept the bearings and seals. The outer end of the stub is threaded to take a large spanner nut, and notched for a lock ring.

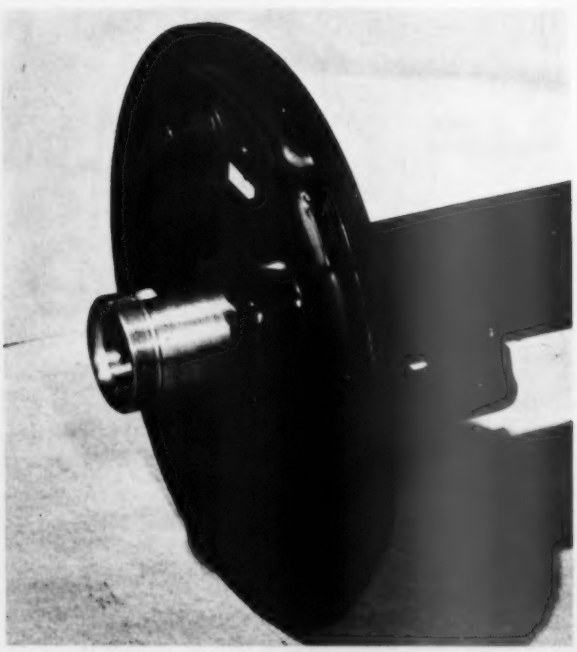
For use with heavy duty floater assembly, housing must be cut off, chamfered for weld bead and then bored to a diameter that allows press fit for stub. Work shown in progress in photo.

Floater stub welds to stock axle housing. New unit for drag cars will bolt to existing flange, need axle flange machining.

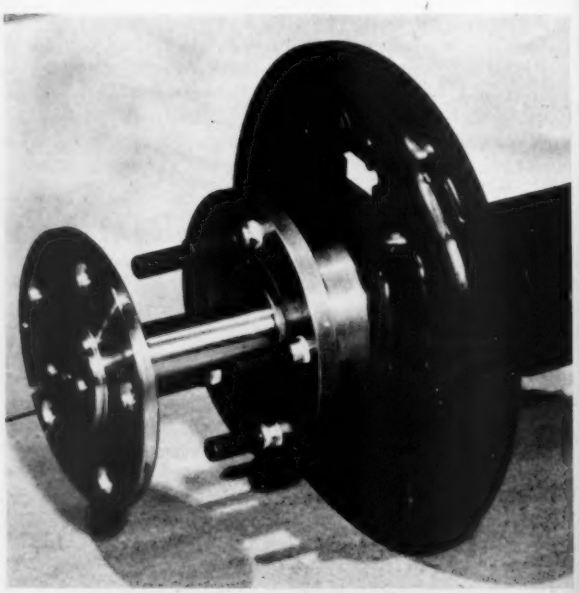
The floater hub is made from cast steel and machined on the inner bore to accept bearings and seals. The outer end of the hub has a flange and is bored to match the bolt pattern of the axle that is to be used. Long studs are pressed into this hub and when in the car extend through the axle flange, the brake drum and the wheel.

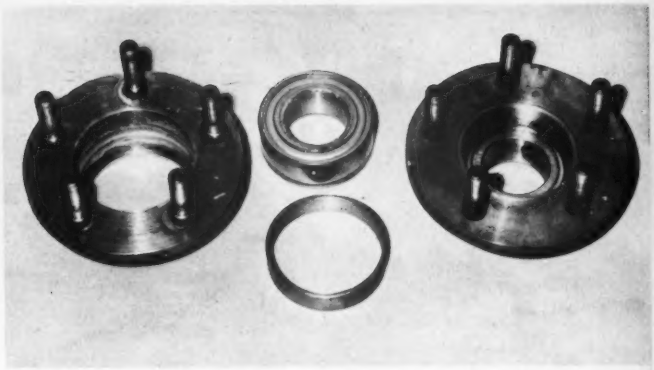
Stock heavy-duty rear axles of the particular model car being fitted are machined to work with the floater hub. First, studs are pressed out, the stock bearing and retainer plate are removed, then the shoulder against which the stock bearing seats is trimmed off. The inner face of the axle is cut down to a flat surface so that it will draw up flat against the hub. Next, the stud holes are reamed slightly larger so that they will slip over the longer studs in the floater hub. When this axle is in place, the machined surfaces of the hub and the back side of the axle flange mate to prevent grease from leaking inside the brake drum.

You will notice that there are actually two different types of bearings used in the hubs in the pictures on these pages. One unit uses a sealed double-row ball bearing while the other uses two individual tapered roller bearings. The double-row ball bearing was originally used and worked out perfectly well at such places as the Daytona Speedway and Atlanta Speedway but H & M later found that by using the



Above—With heavy duty floater unit, brake backing plates must be reworked with center cut out so that the rear of the hub can stick through. Bore in stub is slightly larger than the shaft. Right, complete unit looks like this. Extra long studs in hub pass through axle flange, brake drum and wheel. Axle can be removed for towing by using spacer plate of proper thickness.



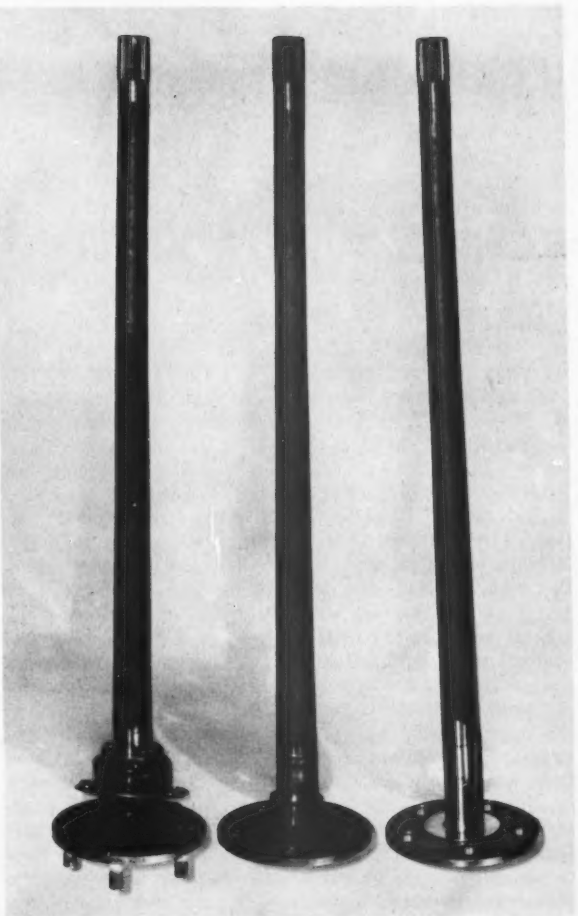


Floater stub and hub used with double-row sealed ball bearings are shown in various stages of manufacture. Stub, top, is machined from chrome moly steel, the hub from steel casting.

Above right: From left, the hub, sealed double-row ball bearings, spacer and the assembled hub are shown. The modified back side of axle flange holds spacer and bearing in place.

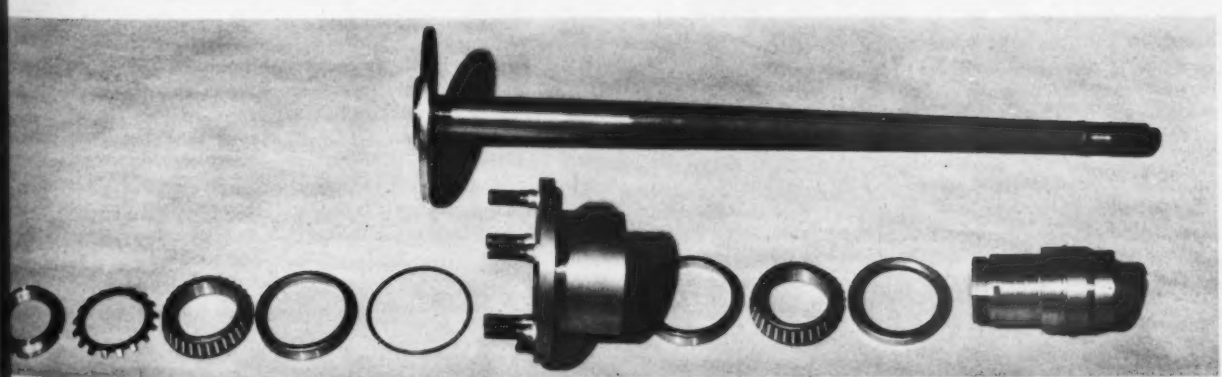
tapered rollers with a larger inner diameter, they could increase the wall thickness on the floater stub and increase strength. While the first hub can be used with stock brake backing plates, the larger hub requires that the plate be reworked. Although the latter unit is more expensive due to extra machining operations, it is a much stronger unit and rather than make two separate models, all future units for race tracks will be of the heavier variety.

The original unit which uses the sealed double-row ball bearings has not been discarded though, it is being redesigned to work as a bolt-on item for drag strip cars. Whereas the heavy-duty track unit requires the car owner to ship a housing and two axles to H & M for modifications, the bolt-on lighter unit will be exactly as the name implies. One exception is that the back side of the axle flanges will have to be machined to match the floater hub. Prices are not firm yet on the bolt-on unit but they should be approximately $150, about half that required for reworking a housing for track use. This is not a cheap unit in price or workmanship but when installed, should be a valuable addition to any performance automobile. **RAY BROCK**



Right—Axles are shown in three stages: stock, minus bearing and studs, and completely reworked to fit the floater hub.

Holman & Moody heavy-duty "floater" has chrome moly stub which is welded to axle housing, cast steel hub which carries two tapered roller bearings. Large nut holds hub to the stub.



HIGH POWERED RACING FUELS

THERE have been fewer things that have been more misunderstood in the automotive racing world than the stuff that goes in the fuel tank of a car. When the internal combustion engine was first invented and developed, it was almost an afterthought as to which type of fuel would be used. Engine builders grabbed the first thing that seemed to be handy, and which the petroleum industry was throwing away at the time. The pattern was set. A liquid it was to be, that was easy to store and easily atomized by a carburetor.

As the sport of auto racing grew, people began to investigate the possibilities of various kinds of fuels other than petroleum distillates. From this experimentation it has become almost an axiom that where the fuel type is unrestricted, gasoline is not used at all. For example, the contestants at the Indianapolis 500, all Land Speed Record attempts (Dr. Ostich's jet car uses jet fuel) as well as the present holder of the LSR, the past and present holders of the standing Kilometer acceleration record, all used substances other than gasoline.

Drag racing in its earliest days had no restrictions on the fuel used, but as the sport developed and classes were established, it was recognized that gasoline was running under a handicap and separate classes were set up, one for gasoline and one for all other types of fuels.

Early in 1957 the organized drag racing sport, spearheaded by the National Hot Rod Association, decided to restrict its racing to pump gasoline alone. There were many factors involved in this decision, the most important ones being: safety, the high cost of these exotic fuels, and the lack of mechanical perfection in most fuel burning machines of that day. Prior to this time, it was quite evident that the number of accidents among fuel burning cars was increasing greatly in comparison with the actual number of this type of car. The accidents were attributed to two causes. The first was the poor handling characteristics of these machines, especially off the line, due mostly to chassis design and poor weight distribution. The second and probably most important cause was the volatility and extreme flammability of fuels. When gasoline burns in air, it does so with a visible flame but when most other fuels burn, the flames are so nearly invisible that many times the driver could be seriously burned while no one was aware of it.

The "gasoline only" ruling actually affected very few people as gasoline has always been the most popular fuel. This ruling has since proved to be extremely beneficial in encouraging the mechanical perfection of both chassis and engine, and now both fuel burning and gasoline powered cars are turning similar times in the quarter mile. Up until 1957 it was common practice by many contestants to use fuel in *hopes* that it would solve *all* of their problems; *which it seldom did.* Many mechanical innovations were neglected due to this attitude, as fuel was commonly believed to be a cure-all. The fact that this is not true has been borne out by the past few years experience in drag racing, but there seems to be no doubt as to the good performance of the fuel burning race cars in the hands of *experienced* builders and tuners. Because of this and other reasons, organizations such as the Bonneville Nationals have elected to allow the use of any fuels, but have also made strict differentiation between gasoline and fuel classes.

Before looking at some of the reasons for the differences between fuels, let us look at the physical properties that a substance must have in order to qualify for use as an automotive fuel. First, it should be a liquid at ordinary temperature (anything can be liquefied if cooled and pressurized enough, but who wants a giant thermos bottle for a fuel tank?) so that the existing carburetion systems can be readily adapted for it. Since the function of the carburetion system is to vaporize the liquid, gases may be used if special devices to regulate amount, etc., are utilized. Solids can also be used, but only if they can be atomized. This can be done by dissolving them in a suitable solvent, or less satisfactorily, to make a slurry (a solid suspended in a liquid) which can be metered.

Second, the fuel must be stable in storage and also when mixed with other parts of the fuel mixture so that the release of energy (explosion) takes place in the cylinder, not in the fuel tank, lines or carburetor.

A third property that has been neglected by many and has affected quite a few is the toxicity or poisonous nature of the fuel and/or its combustion products. This can be very dangerous as people have been seriously injured by *merely smelling* some of these fuels. This is one place where what you don't know can hurt you.

A fourth consideration is strictly relative, and that is cost. Some fuels sell for 30 cents a gallon and some sell for as much as 100 dollars a gallon. And the worst part is that the more expensive the fuel, the more fuel is needed.

Remembering these requirements, let us take a closer look at the characteristics of some fuels and how they compare to the requirements of an engine that will produce maximum horsepower. To begin, a few relevant facts need to be pointed out. Without going into this too deeply, let us just state that the horsepower of an engine is directly related to the weight of the charge in the cylinder. This weight can be increased in a number of ways, the easiest of which is to increase the size of the cylinder (no substitute for cubic inches school). Another way is to increase the volumetric efficiency, or make it breathe better. This can be accomplished by the usual hopping-up techniques of bigger valves, porting, multiple carburetion or fuel injection, and the most recently exploited system of "ram" tuning. These and other paths toward increasing the charge in the cylinders are used in all engines set up for racing regardless of the type of fuel used. A second consideration that we will point out is the basic requirements needed for combustion of any kind, whether it be a bonfire or in a rocket engine. Since we are dealing with an internal combustion engine these same requirements must be met. Three things are needed: a fuel (called a reducing agent by a chemist), an oxidizer to react

with (burn) the fuel, and some energy to initiate the reaction. The latter is usually accomplished by the ignition system (since we aren't discussing diesel engines here), by means of the sparkplug fired by either a coil or magneto.

Let us explore the first two more fully. For the time being we shall take care of the oxidizer by letting it be the oxygen of the atmosphere. It is then more or less constant and consists of about 21% of the air, 79% being nitrogen and other gases which will not react under the combustion conditions found in the cylinders of an engine. There are small amounts of water vapor and carbon dioxide also in the air, but these will be assumed constant along with the oxidizer in the cylinder combustion process. Such is the usual case in unblown, gasoline burning engines in any stage of modification. This will enable us to concentrate upon the fuel and its role in combustion.

The composition of the different fuels varies, but the most commonly used ones contain carbon and hydrogen as the main constituents. Going back to basic chemistry, we recall that when equivalent weights of hydrogen and carbon are burned to give water and carbon dioxide respectively, the hydrogen gives off much more heat. When carbon burns to the intermediate oxidation product carbon monoxide, it requires less oxygen for the same amount of carbon, a fact which helps to explain why an engine will produce more *power* when the combustion product is mainly carbon monoxide (excluding the water vapor) in spite of the fact that the production of carbon dioxide gives more *energy*. Also the actual path of combustion is such that hydrogen burns first and if there isn't enough oxygen to go around, the carbon gets left out. Thus we get carbon deposits built up in the combustion chamber. This would indicate that if we could use pure hydrogen for a fuel, we would have the ultimate end, and that is just about the case. But there are a few factors such as low temperatures and high pressures needed to liquefy hydrogen that prevent even the U.S. government from using it as a rocket fuel . . . and they have millions of dollars to play with so it seems that we must look elsewhere.

Gasoline is the most used automotive fuel and deserves a bit of attention as to how it achieves this distinction and why it doesn't qualify as the ultimate fuel for a high horsepower engine. If Uncle Sam spends millions on rocket fuel research, the petroleum industry spends that much or more on its main product, gasoline. They have constantly improved the quality of refining, cracking, etc., until gasoline of better than 100 octane is available for little more than it was 30 years ago, when octane was considerably lower.

Let us explore the composition (Fig. 1) of the gasoline molecule. We find that each tiny particle contains from seven to eleven carbon atoms each, with a little more than twice as many hydrogen atoms per molecule. Since the amount of oxidizer (oxygen in the air) is constant and hydrogen burns much more readily than carbon, these large molecules (and they are large by the standards we shall set) must be metered very closely into an engine, which is accomplished by means of the main metering jets in the carburetor or injector. As is well known, a variation of 2- or 3-thousandths of an inch in these jets can make quite a difference in performance. With gasoline it doesn't take too many thousandths of an inch increase to make it too rich, which means that even though the charge is heavier in the cylinder there isn't enough oxygen to burn the molecules that are present, which results in a loss of power.

The obvious step would be to find a fuel with smaller molecules still consisting of carbon and hydrogen, so we can put more of them in and have them burn using the same amount of available oxygen. By this means we have

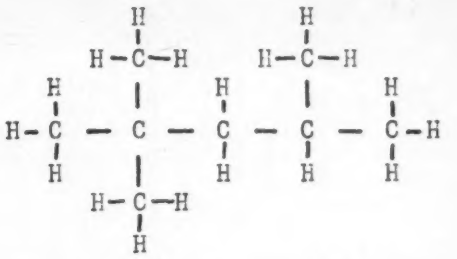


Fig. 1 A typical gasoline molecule "iso octane"

Relatively large gasoline molecules must be metered extremely closely by main metering jets in the carburetors or the injectors.

Fig. 2 Methanol "alky"

Fig. 3 Nitromethane

Small molecules are the stock-in-trade of methanol, but nitromethane also has small molecules, and is most used of all fuels.

effectively increased the ratio of hydrogen to carbon, which gives more heat energy when burned. Such is the case when methanol (alcohol) is used. Its molecule (Fig. 2) consists of one carbon and four hydrogens (and one oxygen which doesn't influence the combustion process to a very large extent, but does give methanol the property of being a liquid at ordinary temperatures). With this smaller particle we can meter it richer than we were able to meter gasoline and still expect complete combustion. This puts more in the cylinder, thus increasing the charge; the very effect we were looking for. There is still a point of being too rich or having too many molecules for the supply of oxygen available, and since methanol has the smallest molecule of the common fuels, we are as far as we can go in the direction of size.

The next step is to do something about the oxidizer. The simplest solution is to add oxygen, and this has been done with some success, but it requires elaborate metering set-ups and if oxygen happens to be in excess at any time it tends to burn almost anything available such as pistons, valves, etc. Another way to increase the oxidizer is by supercharging, which increases both fuel and oxidizer. This is a subject for an entire article by itself, so we will merely mention it in passing. The third way is by adding solutions of chemicals known to the chemist as oxidizing agents. These are chemicals that are not too stable and tend to give up oxygen without much trouble under the influence of heat and pressure, especially in the presence of a reducing agent (fuel). The most commonly used compounds are the nitrogen-containing chemicals. These are notably unstable and come apart with ease. (Nitroglycerine is a prime example of this ease!)

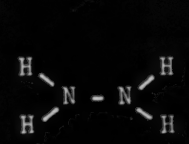
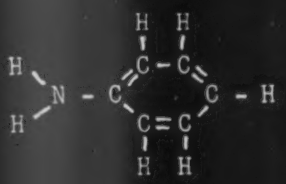


Fig. 4 Hydrazine



Aniline

***Left*—Both hydrazine and aniline are classified as high energy fuels, are stable and somewhat expensive, can be mixed with gas but have affinity toward water. *Below left*—Solid fuels made into "slurries" such as naphthalene, picric acid, and trinitrobenzene have been used but tend to wear out fuel system pieces rather rapidly. *Below*—Hydrogen peroxide and nitric acid act as the oxidizers in potent fuels.**

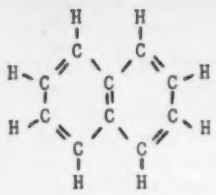
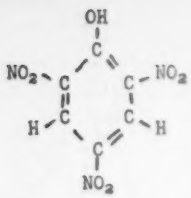
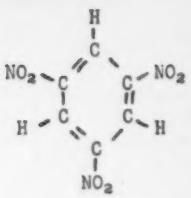


Fig. 5 Naphthalene



Picric Acid



Trinitrobenzene

$HONO_2$

Nitric Acid

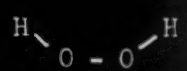


Fig. 6 Hydrogen Peroxide

Since we are looking for a small molecule containing both carbon and hydrogen in addition to nitrogen and oxygen, we find that the one which fits these requirements best is also the most popular, nitromethane (Fig. 3). Its initital use is lost in the confusion of history, but it was considered for a rocket fuel by the U.S. government during the V-2 rocket experiment period. It makes an excellent fuel as it is both oxidizer and fuel combined; that is, it contains more than enough oxygen to burn itself and consequently has some left over to burn whatever else is in the cylinder, be it fuel, pistons, valves, or just the cylinder itself. This characteristic makes it mandatory to keep the mixture on the rich side and enables us to meter it very rich with subsequent increases in charge density and horsepower. The limiting factor for this type of fuel is neither the oxidizer nor the reducing agent as it contains both of these. Indeed, the oxygen of the air has little to do in the case of an engine burning large quantities of nitromethane. The only function of the atmosphere in this case is to push the charge into the cylinders and the engine would operate as well in an atmosphere devoid of oxygen. Then it seems that the more fuel that is put in, the greater the horsepower output. This is true and is only limited by the amount that can be put in. The limiting factor then becomes the energy of the fuel.

The way to greater power from here is applying the same requirements to even higher energy fuels. One place that we can look for these is that area which has been mentioned several times, the U.S. government rocket research programs. Or maybe even better would be the Russian government rocket research, if you can read Russian! These requirements are astonishingly similar to the needs of a racing engine. For our uses we will restrict ourselves to the liquid fuels and leave the solid ones alone. As we stated before, there must be some stability attached to a fuel if it is to do us any good. Also, just because a fuel is used as a rocket fuel doesn't necessarily mean that it is a high energy fuel. For reasons we won't go into now, there is plenty of use for things such as kerosene and other types of low energy fuels. Also remember that dragsters don't use liquid oxygen for an oxidizer either.

Then looking over the field we can pick such possibilities for high energy fuels as hydrazine and its close relative aniline (Fig. 4). Both are liquids, stable and quite expensive. The saving grace here is that both can be mixed with lower energy fuels such as gasoline, methanol, nitromethane, etc., to give large proportionate increases in fuel energy. Both are somewhat toxic and have quite an affinity for water. Since water in an engine belongs either in the cooling system or as a product of combustion, this is not a desirable property. There are people who have tried a small amount of water as a cooling agent directly in the fuel but this seems to do curious things to cylinders and pistons. The effect is that of termites in wood; many little "worm holes" completely through the piston.

Boron fuels are getting a lot of attention these days due to their light weight and high energy. The main objections to these as automotive fuels are their tremendous cost and also that most are gases, not liquids. Pentaborane, a stable liquid, might be considered, although the main combustion product is boric oxide, a solid, fusible material which would be messy in an engine.

There are also solid fuels which are usually used in solution. Slurries tend to wear fuel lines, fuel valves and metering devices too rapidly to be much good. Also the problem of mixture control is difficult with these. Naphthalene has been used to good advantage as have others such as picric acid, trinitrobenzene, and related compounds (Fig. 5).

There are oxidizers to be considered here also (Fig 6). Hydrogen peroxide (90 percent) has been used in small quantities as an additive. This releases much oxygen for its relative weight, but is dangerous to handle due to its instability. Concentrated nitric acid can be used; in fact a small amount added to nitromethane "ages" the nitro in a hurry. This forms a small amount of dinitromethane which is quite a high energy fuel. Various solid oxidizers have been used in solution. Potassium permanganate colors the

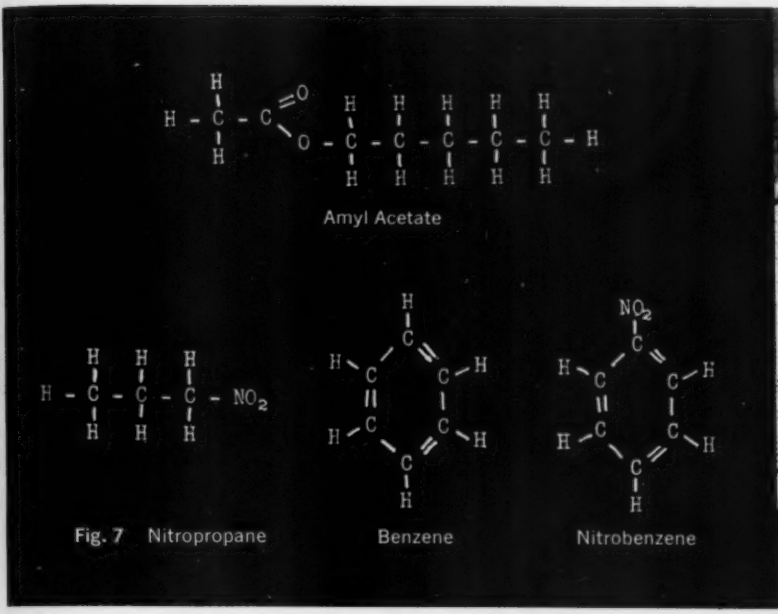
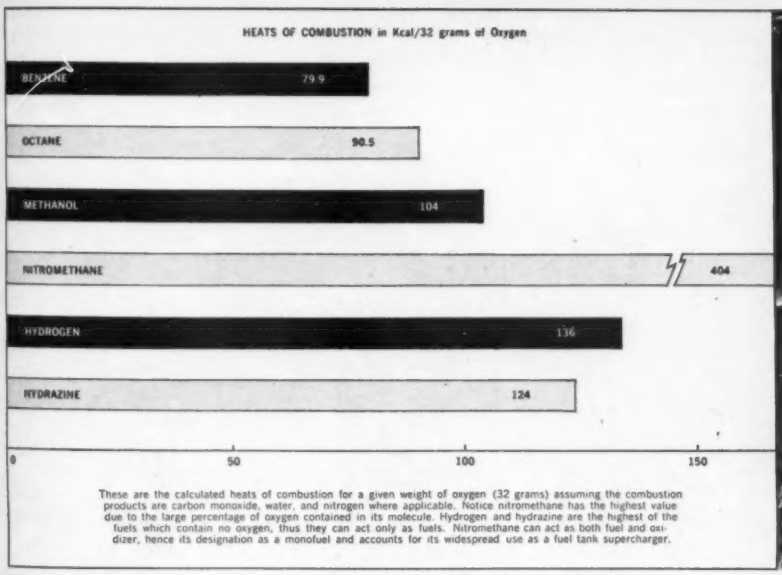


Fig. 7


HEATS OF COMBUSTION in Kcal/32 grams of Oxygen
BENZENE 79.9
OCTANE 90.5
METHANOL 104
NITROMETHANE 404
HYDROGEN 136
HYDRAZINE 124
0
50
100
150
These are the calculated heats of combustion for a given weight of oxygen (32 grams) assuming the combustion products are carbon monoxide, water, and nitrogen where applicable. Notice nitromethane has the highest value due to the large percentage of oxygen contained in its molecule. Hydrogen and hydrazine are the highest of the fuels which contain no oxygen, thus they can act only as fuels. Nitromethane can act as both fuel and oxidizer, hence its designation as a monofuel and accounts for its widespread use as a fuel tank supercharger.

fuel a beautiful purple and seems to work well if the system is flushed immediately, a must with all of the fuels mentioned.

Such fuels pack tremendous amounts of energy, and the mixing and metering of these fuels into an engine is a trial and error process, with even the most sturdy engines often not standing up to the treatment. Before fuels are mixed together haphazardly, a decision should be made regarding the type of fuel base to use. Usually methanol ("alky") is used, even though benzene (benzol) is quite popular. To this base is added varying proportions of nitromethane, usually in easily measured increments by volume. Thus "50 percent" has come to mean an equal volume of base fuel

***Top left*—One of the problems in using fuels comes in mixing. Blenders, known as esters, tend to mix with desired fuels and most usually give off a sweet or fruity smell. *Top right*—Indianapolis cars run on methanol, can lose their innards from too wild a fuel mixture. *Above right*—Tony Capanna's Dodge V8 built for Indy 500 in 1955 lost main bearing webs in practice. Engine ran on a nitro blend, which was attributed by some for failure, but Tony felt a lubrication problem was the cause.**

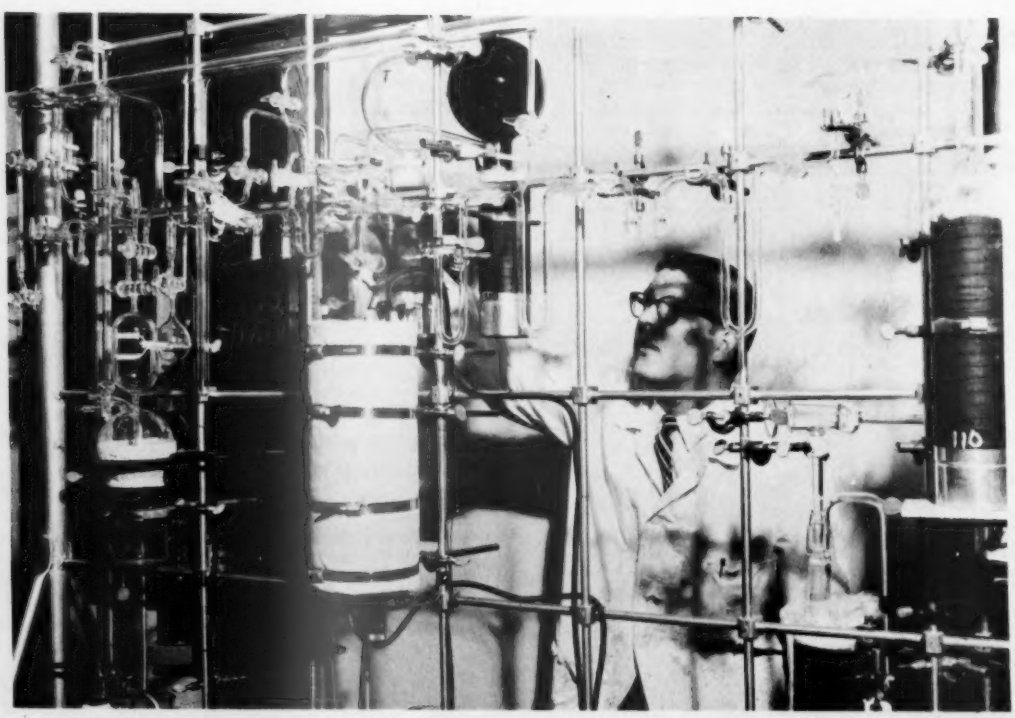
mixed with an equal volume of nitromethane. Because the manufacturers put a small amount of benzene in nitromethane to stabilize it in shipping and storage, it is sometimes advantageous to use benzene as the base. However, when benzene is used as the base, nitrobenzene is the natural choice for an additive (If nitrobenzene gets to be more than a 50 percent ratio, which is then the additive?). Since benzene is added as a stabilizer, it becomes rather difficult to run 100 percent nitromethane when only about 98 percent is available. When this point has been reached, the next thing to do is add one of the high energy fuels mentioned. These are usually used in small percentages such as from 2 to 5 percent, with subsequent bigger jetting to insure a rich enough mixture. Engines set up with carburetion must have the carburetors modified greatly to meet fuel requirements. Dump tubes should be at least .210-inch inside diameter and, when using these mixtures, the engine should not be allowed to run slowly as this allows the pressure differential to become too small on either side of the carburetor venturi, resulting in a lean mixture which is highly destructive. Since this problem does not occur with the use of constant flow fuel injectors, injection is probably the best solution. Fuels are a very expensive way to power at best, and there are many times when mechanical perfection doesn't seem to count for much, if the engine will only hold together. [*One dragster went through more than 30 DeSoto and several Chrysler engines in approximately 3 years time. This machine cost right on $14,000 for maintenance alone, which shows dramatically why fuel is an expensive proposition—ED*].

We have skipped over many less energetic fuels but some of them are in use due merely to more desirable physical or chemical properties that enable them to be used in spite of the fact that higher energy fuels are available.

One of the problems of using fuels comes in mixing. Even though nitromethane won't mix with gasoline, nitropropane will, or a solvent can be found that will mix with both constituents as a kind of blender. The compounds normally in use as blenders are known as esters (Fig. 7), and most of them have a sweet or fruity odor. The most common one is amyl acetate. A small amount will cause oil and water to mix, thus it has a great variety of uses in mixing fuels. So we see that all of these fuels have one basic thing in common, and that is to get more fuel into the cylinder under conditions wherein it will burn.

Because of the high initial cost of these fuels, the inherent lack of safety in using them, and the high rate of consumption (it isn't uncommon for a dragster to get one quarter-mile run on something over a gallon of fuel), several organizations across the country have restricted their racing to pump gasoline only. This indicates that for the fairness of the sport, means must be devised to run tests on the fuels being used. This is not nearly as complicated as it first seems, due to the fact that all fuels can be placed into a few groups and quite simple definitive tests can be run. Since most of these organizations are run by people that have been in racing since the "year one," the possibility of unknown fuels or combinations is very unlikely and variations of the test procedures would show cheaters up rapidly. Also, new procedures are constantly being worked out with that aim in mind. Apparently it would take a very brave person to attempt to cheat with fuel, for it is almost a certainty that he won't get away with it for long and there are a number of people who can attest to the efficiency of these fuel test procedures. There are plenty of legal places to try out a new fuel mixture; the long white stretches of salt at Bonneville for one. And, of course, there's always the space agencies!

DEAN HILL



Dean Hill with portion of equipment used for fuel analysis and experimentation at Michigan State University Department of Chemistry. Experimentation and use of super fuels should only be used by experienced persons.

PICKING THE RIGHT CLASS

SOME guys are born speed mechanics. They can take a car that falls in any class at the drag strip, and with a little special engineering, tuning, construction and driving—and usually with a minimum of green stuff—can win trophies.

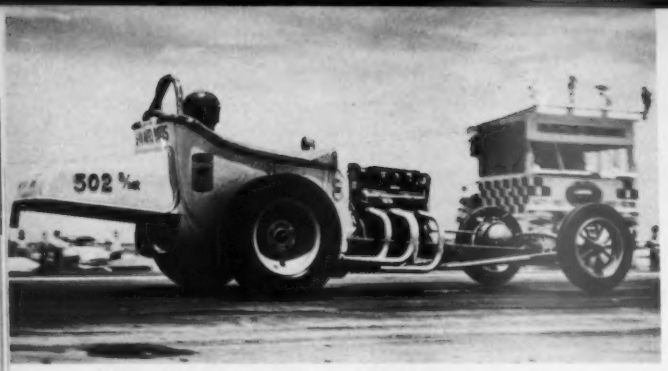
These fellows are few and far between. For every one of them there are a hundred of us "average guys" who have to use every means possible (this side of outright cheating, I hope!) to take home two or three trophies a *season*. It would take a whole book to even begin to list the possible speed secrets that might help us here. But there's one secret that comes before all the rest—(and it may be the most important of all). That is *selecting the right class to run in*. Class selection, and the buying or building of a car to fit a selected class, can be the difference between jumping into a hornets' nest of furious machinery that leaves you in the dust, or getting into those close fender-to-fender duels that make drag racing fun even if you don't win all the time.

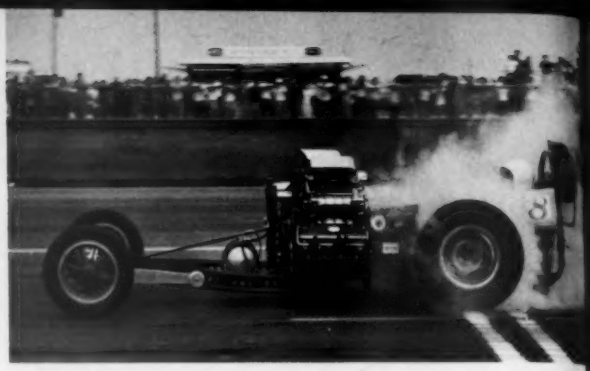
There is a beautiful symmetry and logic about the National Hot Rod Association's drag strip classification system that every drag enthusiast should thoroughly understand.

In the first place, the performance of any car on the quarter-mile drag strip is going to depend heavily on just three factors: (1) Its horsepower, (2) its weight, and (3) the front-rear weight distribution. The weight/hp ratio is the dominant factor in determining the average rate of acceleration in the speed range above 30 or 40 mph. The relative front-rear weight distribution is the major factor in the car's *traction*, which determines the all-important acceleration off the line for the first couple hundred feet. At these low speeds the proportion of weight on the rear wheels—which establishes the maximum thrust-weight relationship (and, in turn, the acceleration)—is more important than the weight/hp ratio. So the NHRA officials took these three fundamental factors and built a whole class system on them.

In the stock car division they were able to use a logical weight/hp ratio for classifications by merely dividing the published shipping weight of the particular car model (including any added weight for optional V8 engine and/or automatic transmission) by the advertised hp rating. This obviously isn't a *perfect* classifying system because some engine models put out more *true* horsepower at the clutch, in relation to advertised hp, than others; but it has the important advantage of using widely available reference data for figuring classes—and it's been working out very nicely for several years, with minor changes in class divisions now and then.

Now the rulesmakers had an entirely different problem with the modified divisions. Here there are no published power and weight figures to fall back on. The fellows hop the engines up out of all proportion to their original ratings





Variations in car weight, engine position and engine displacement all help decide a competitor's class. Modified roadster (left) can be steered from back porch if necessary for proper weight disposition, while dragster can follow book to the letter.

(or in many cases their honest output more closely *approaches* the advertised rating!), and bodies and chassis are modified to entirely different weight figures. Obvious answer: Weigh the cars right at the strip, and divide by the cubic inches of piston displacement. Classify on a basis of *pounds per cubic inch,* with super-chargers moved up one class. This puts it squarely up to the car builder to get the maximum hp output *per cubic inch.*

But now we get into the subtle part of this classification theory. Forget body rules for the moment. Now note carefully the NHRA modified classes are divided into three broad categories—based roughly on potential front-rear weight distribution They control this factor by controlling the *position of the engine on the wheelbase,* which is the major moveable mass on the car. In the Gas Coupe/Sedan and Street Roadster divisions the engine must be placed in the stock position. In the Altered Coupe/Sedan and Roadster divisions it may be set back 25 percent of the wheelbase (measured from the centerline of the front spindle to the front spark plug hole). The Competition Coupe/Sedan, Modified Roadster and Dragster divisions permit *any* engine location, providing not less than 30 percent of the total weight is on the front wheels with driver seated.

The body requirements in these various divisions (maximum height reduction, fender requirements, etc.,) are of importance to the builder, as we'll see later; but they actually have a relatively minor effect on performance. That is, note the similarity in performance between coupes and roadsters in a given category. Even dragsters don't seem to have a big performance edge over coupes and roadsters, given equal weight ratios. This is undoubtedly because the open wheels and tires make up such a large proportion of the total wind resistance on these machines that the tiny dragster body is no big advantage. Anyway, it's pretty hard to beat this system of classifying on weight, cubic inches and engine position. Really clever.

Incidentally, there has been no need to consider this weight distribution factor in the stock and sports car divisions because of the similarity of all competing cars. But I'm wondering about the effect of Chevrolet's new rear-engine Corvair, which has 62 percent of its static weight on the rear wheels (compared with about 45 for front-engine stock cars). I predict if they get some gears and horsepower in that thing it will jump its class competition so far off the line that the NHRA boys will have to start juggling some stock classes!

PICKING A DIVISION

The term "division" is not used in most drag strip rulebooks; but I like to use it to distinguish between the various body classifications. That is, Gas Coupe/Sedan, Roadster, Competition Coupe/Sedan, etc., are *divisions*—then each division is split up into *classes* based on pounds per cubic inch. The potential drag strip competitor must first pick his division, then pick a specific class in that division.

Actually there's one all-important rule to keep in mind when picking a division: *take advantage of all the allowances in your chosen division.* Perhaps a couple of examples will illustrate here. Let's say we have a '32 Ford coupe—a genuine "hot rod"—that's probably been knocking around the area for eight years or so. It's only mildly channeled, not chopped, but the fenders are stripped and there is only partial interior upholstery. During its checkered career the car has picked up a late V8 overhead in stock position.

Believe it or not, this car is a freak as far as drag strip classes are concerned. We just don't have a specific division for it. As it stands it would run in Altered Coupe/Sedan. This division permits fender removal, gutted interior, maximum body height reduction of 10 inches by chopping and channeling. But it also permits the engine to be set back 25 percent of the wheelbase . . . and our '32 has a big 700-pound overhead 'way up in front. It would be at a hopeless disadvantage against all-out Altereds with more weight on the rear wheels. Almost regardless of how much horsepower we had for acceleration at the end of the quarter, they would gain so many car lengths off the line we could never catch 'em.

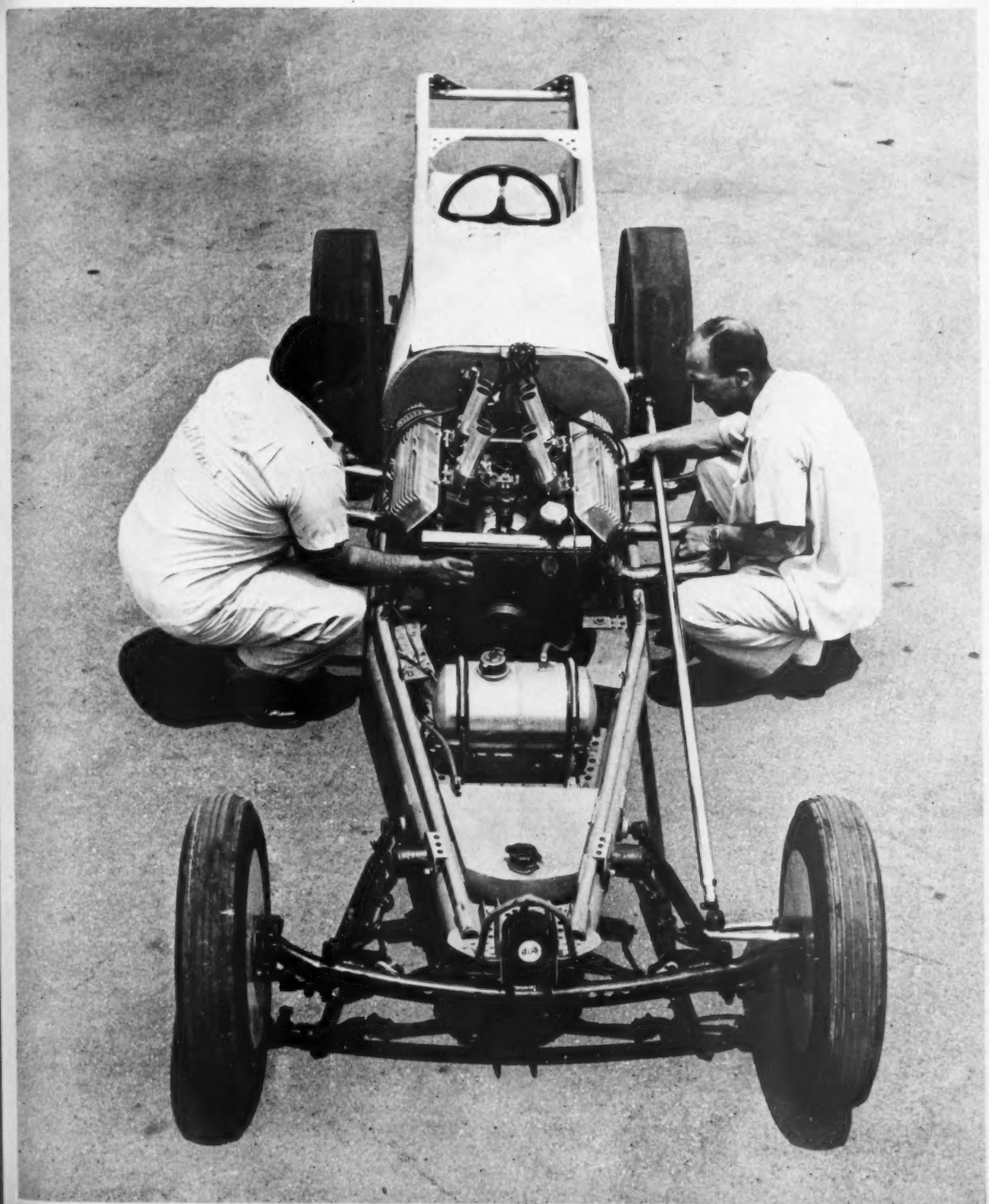
No, I think in this case we would be better off converting this "hot rod" into a "gas coupe" by replacing the fenders and full upholstery. This division requires the full body, fenders, upholstery, stock engine placement, and permits a maximum height reduction of four inches. Our '32 should qualify okay here—*if* the body hasn't been lowered too much. One thing is sure: It would be at a great disadvantage in the Altered division as it stands. We should either rebuild the car completely for this division by moving the engine back 25 percent, or go the other way toward the Gas/Coupe Sedan division by replacing fenders and upholstery—even if it might mean a major job of "de-channeling" the body to raise it within the four-inch rule. We *must* take advantage of those major allowances in our chosen division.

Another typical problem along this line is like the boy with, say, a '54 Olds 88 sedan with three two-throat carbs and a reground cam. This car is modified, of course, and would fall in the Gas Coupe/Sedan division—and in the red-hot Class C at that. What chance would it have here? We'd have to run against a variety of iron, blown and unblown, that can turn e.t.'s in the 14's and 13's—sometimes in the 12's! Only a miracle would get us a trophy. Obviously we would be better off going back to a stock cam and four-barrel carb, and running in the C/Stock class. A well-tuned '54 Olds would have a good chance here, especially if your strip runs the optional automatic transmission class.

It all goes back to what I said earlier about taking advantage of all the rule *allowances* in your division. When the

rules allow the engine to be set back in the chassis, *set it back*—or get out of that division. When the rules allow the engine modifications, *go all the way*—or go back to stock. Half measures can't win trophies consistently. The only condition that would alter this principle would be that competition in a particular division might not be keen in your particular section of the country, so you could win fairly consistently with second-rate equipment. In the far west and southwest, the cradle of the sport, competition is rough in *all* divisions; you have to have a top car and driver to win week after week in these areas. In the midwest and south competition tends to be spotty in some altered and competition coupe and roadster classes. It's even more so on the east coast. Many times you'll be the only car in your class with anything hotter than a gas coupe. The fact that the drag sport is newer in these areas, and the season shorter,

Front-to-rear weight distribution is a major factor in proper control of the car's traction.



Despite heated contests, hot rodders are known for friendly rivalry. Here two drag racers discuss ways to help each other.

has a lot to do with this. Anyway, in this atmosphere, it's entirely possible to really make hay with that "genuine hot rod" with the 700-pound overhead up front. More power to you.

PICKING A CLASS

After you've picked a promising division to hit you still have some choice left in selecting a class in that division, based on pounds per cubic inch. You can either adjust your piston displacement when you plan the engine hop-up job, or you can adjust car weight—either by lightening or increasing car weight with ballast. The new NHRA rules limit removeable ballast (must be securely fastened in) to 100 lbs., but you can use more if it is solidly welded in or built in as part of the car structure—and providing the tech crew passes your installation as safe in case of a crash.

So how can we get cute with the pounds-per-cubic-inch class system? We can't if we stick with Class A. In each division Class A is for all cars from *zero* lbs. per cubic inch up to some maximum figure, like say 6.59 lbs./cu. in. You're never really "safe" in A. There's always the chance you'll run into somebody with a better lbs./cu. in. ratio than you have, regardless of how big your engine is or how carefully you've chopped weight. It's a good class for beginners to avoid. (An obscure advantage is that you don't have to worry about tearing down your engine for protests because there is no specific cubic inch limit.)

We *can* get cute in the lower classes. Think about it a second. Theoretically optimum quarter-mile performance will be achieved, within a given lbs./cu. in. limit, by using the *biggest* practical engine—and then planting a huge amount of ballast *in the rear of the body* to make up the necessary weight. This will increase the relative proportion of weight on the rear wheels, but we'll still have the lowest lbs./cu. in. ratio allowed by the rules. There's only one bug in all this: We must always strive for the maximum hp output *per cubic inch*—and it is a well-established axiom of engine design that the potential hp/cu. in. goes up as the piston displacement *comes down*. Smaller cylinders have less friction, run cooler, can use more compression, can get more relative valve area, can turn higher rpm's. Certainly a well-built 265-inch Chev V8 can put out more per inch at 7500 rpm than a bored and stroked 339-incher at 6000.

So we've got to compromise. Moderate cubic inches—moderate ballast. Let's see how it works . . .

Let's consider the case of this 265-cubic inch Chev engine in the Gas Coupe/Sedan division. Most late Chev sedans with this engine run in Class D, which covers from 13.00 to 14.59 lbs./cu. in. We can figure the minimum car weight with this engine by multiplying inches by the lowest lbs./

cu. in. figure, or 13.00 in this case. This would be 13.00 × 265 = 3445 lbs. Very little ballast would be required in the lightest '55 or '56 Chev to make this. But now let's go on. How about trying for Class E? The lower limit here is 14.60 lbs./cu. in., and minimum car weight would be 14.60 × 265 = 3869 lbs. A late Chev could make this pretty easy, or an early convertible; but the 1955-57 sedan would require 300 or 400 lbs. of ballast to make the weight. This is a practical amount to shoot for, and it will do wonders for dig off the line (providing you have gears to take advantage of it). Meanwhile the 265-inch engine, properly modified, has a better chance of exceeding 1 hp per inch than a 283 or larger bored and stroked version. I will say here that any amount of ballast beyond 400 or 500 lbs. is pretty hard to install permanently in the rear of a car, and do it safely—so better discuss any really wild plans with the drag strip tech crew before proceeding.

We have similar problems in the hotter classes. Let's say we want to build up a '32 Ford coupe for the Altered Coupe/ Sedan division with a 312-cubic inch Thunderbird engine. Let's say the basic car (modified) weighs 2250 lbs. This gives us a lbs./cu. in. figure of 7.2—right smack in the middle of B/Altered. This will never do. We're not taking full advantage of our class allowance, which in this case is 6.60 lbs./cu. in. (the low limit of B/Altered). We can do one of three things: (1) We could lighten the car to 312 × 6.60 = 2060 lbs.; (2) we can bore and stroke the engine to around 341 cubic inches—obtained by dividing 2250 by 6.60; or (3) we can go down into C/Altered by ballasting to 312 × 8.60 = 2683 lbs. The latter would mean another 433 lbs. toward the rear of the car, and would help traction immensely. This might be the best bet of all. Anyway, you get the idea on this business of ballasting for a class. Take advantage of your allowance.

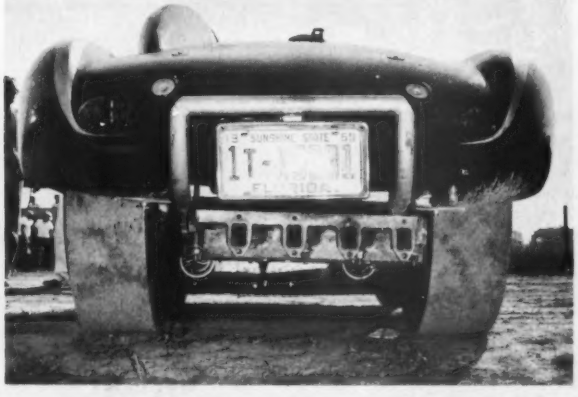
In conclusion, a word about blowers. Let's face it, men—they've done took over. There have been a few class winners at the NHRA Nationals recently that are unblown; but it looks like there will be fewer in the future. Under the present system, where a blown engine moves up one class, the blower is the answer. If you're a good speed mechanic and can afford it, there's no reason why you can't win with a blower, too. But if for some reason you *can't* use a blower, either because of lack of money or know-how, the only way you can stay away from blown competition is to design into the highest available lbs./cu. in. class in your chosen division. Since blowers move up one class, there won't be any in the "bottom" class. I can't suggest strongly enough that you do this. You'll have to accept a lower acceleration level in your competition machine because of the higher lbs./cu. in. ratio; but it beats eating blower dust all the time.

Finally, picking a class in the stock division; actually they're all hotly contested on most strips; none are a push-over. And, let's face it—there's a lot of cheating going on all over the country. In the case of our '54 Olds discussed earlier, when we decided to go back to "stock" the chances are we would've replaced the factory four-barrel carb—but forgotten the reground cam. This is a discouraging aspect of competition in the stock division, for which there is no simple cure. But if you're willing to take your chances among the stock and near-stock, there are several things you might keep in mind when picking a class to buy a car for.

Most important, of course, is to comb the specification sheets for a combination of advertised hp and factory shipping weight that will put you right up near the top of a given class. No use buying a model that's right on the bottom of a class. Most beautiful example of smart model picking I've seen was George Chaltin's choice of a '57 Ford model to win A/Stock at the last Nationals. By searching the weight sheets he found that the heavy Skyliner convertible hardtops could run the supercharged 300 hp '57 Ford engines and just barely fall in A/Stock (whereas all other blown '57 Fords run Super Stock). Result: speeds over 93 mph with e.t.'s in the low 15's—and a relatively easy win at Detroit last fall. There are many other similar possibilities. Other hints: Station wagons make good drag cars because of more rear-end weight. If your strip runs separate automatic transmission classes, keep in mind that the "tight" Hydra-Matic with its fast 1st gear is quick off the line—ditto the three-speed Torque-Flite. Two-speed torque converters are not quite so quick—and you might as well stay home if you've got a multi-turbine Turboglide or Flight-Pitch Dynaflow. And, sorry as we are to bring it up, it's still true that the cheaters don't bite quite so hard in the hotter classes—Super Stock, A and B. Reason: these late high-horsepower engines are already pretty thoroughly hopped by the factory, so "refinements" like hot cams and milled heads don't boost 'em so much as in the lower classes. If you can keep them honest on cubic inches you can handle a hot cam now and then with some smart driving and the right gears.

So you carry the ball from here. Just remember to pick your division and class with as much care as you'd pick a $250 cam kit . . . then come out to the strip with a combination that takes advantage of every *legal* loophole in the rules!

ROGER HUNTINGTON



In street coupe (left), engine position is rather inflexible, but notice cylinder head bolted to the cross-member of sports car.

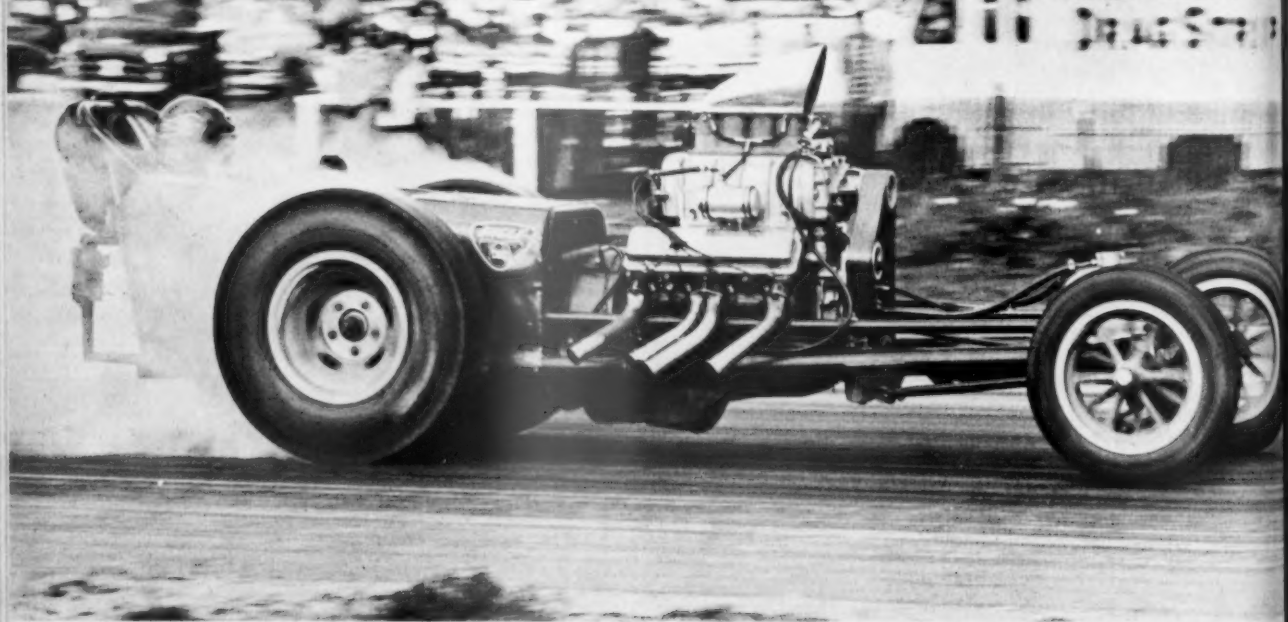
MAGNESIUM WHEELS FOR THE STREET

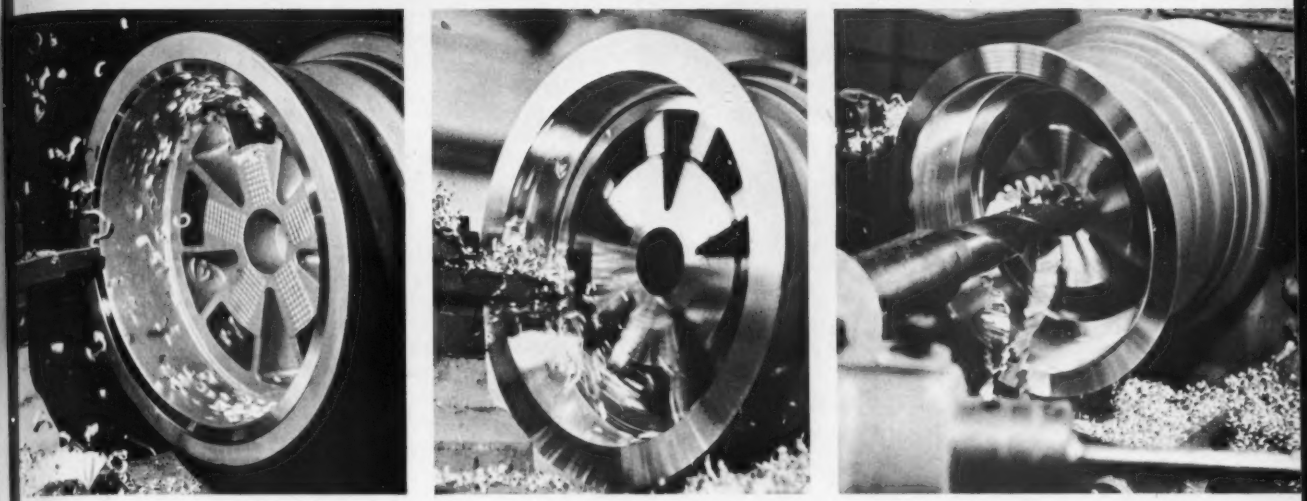
THE MERE fact that a car is equipped with magnesium wheels does not necessarily mean that it will go faster than it might if it were equipped with wheels made from some other material. Most car enthusiasts feel that magnesium is the best metal for making competition or high quality street wheels because of its very favorable weight to strength ratio. It is true that magnesium wheels make it possible to reduce the unsprung weight on a car by a considerable amount. This is, of course, advantageous particularly on high speed track racing equipment. The fact is, however, that there are other advantages to a magnesium wheel which are probably more important in the long run than the weight saving advantage. Probably the most important advantage is the fact that magnesium wheels are made round to very close tolerances. The average automobile steel wheel is out of round a considerable amount. It is not uncommon to find out of roundness of 3/32 of an inch on the average automobile wheel. This out of roundness introduces a series of problems having to do with balance and stability which are virtually impossible to compensate for by adding balance weights. This condition adversely affects tire wear and on a high speed competition machine out of roundness will produce wheel hop at speed which detracts from the desirable handling characteristics in cornering and also from traction characteristics on straight-away acceleration, especially in drag racing. It may, therefore, be said that the greatest advantage of a magnesium wheel today is its roundness and concentricity. The second advantage is the weight saving. The third advantage is the ability to dissipate heat if the mag wheel is properly designed for competition use.

The available choice of magnesium wheels in sizes and shapes, etc., at the present time makes it possible to build a race car with the desired tread geometry and brake drum diameters and tire size for the specific requirements without having to seriously compromise in chassis design because of wheel limitations or interferences.

The most popular sizes of magnesium wheels being used

Miller-Erfman dragster runs American Racing Equipment mag spokes on front, where fine weight balance and lightness are vital.





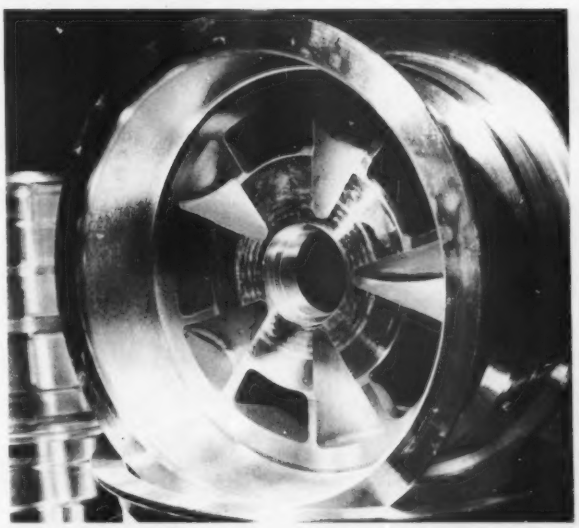
Left to right: First bite on rough casting, turning the face of a magnesium wheel; Machinist performs delicate wheel facing job at American Racing Co.; major step in construction of mag wheels, drilling spindle hole on a turret lathe for utmost in accuracy.

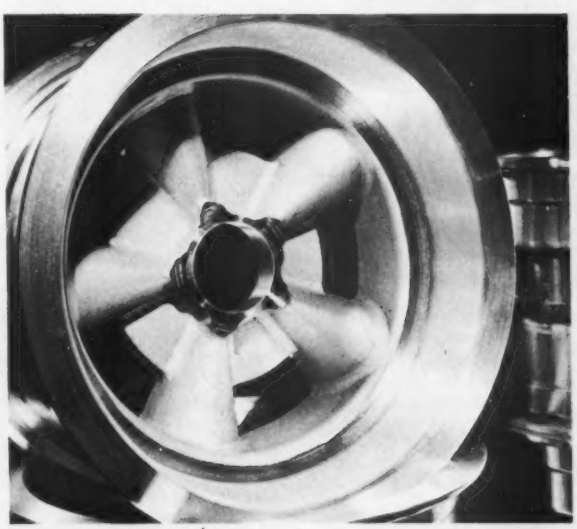
at the present time are those which are designed principally for drag racing or for speedway and dirt track racing. Magnesium wheels for street use and for sports cars have been a minor factor in magnesium wheel production to date. There is, however, an interesting trend at the present time in the direction of applying the advantages of competition wheels to street use and to more extensive use on sports cars for street or strip.

There appears to be good reason for the lack of magnesium wheels on the street to date and until the problem is solved they will probably not be commonplace. The most obvious reason would appear to be the fact that they are comparatively expensive. This would make magnesium wheels for street use a luxury item. A second problem is caused by the characteristics of magnesium which definitely has limitations for street use when it is subjected to the normal abuses of driving. Magnesium must be kept free from nicks, scratches, abrasions, etc., because cracks can originate at these points. For this reason safety requirements at Bonneville and at the Speedway and on aircraft wheels or wherever this metal is used on critical components require that it be subjected to zyglo or even x-ray inspection from time to time. While zyglo is not particularly expensive it is quite inconvenient and, therefore, the average car owner would not care to tie his car up from time to time for the proper inspection and care of magnesium wheels. He would undoubtedly pay considerably more attention to the same nicks if they were in the paint on his car rather than the wheel.

The new Pontiac two-piece aluminum brake drum and wheel combination has met with considerable interest and may foreshadow more of this sort of thing to come from Detroit in the future.

The Cadillac El Dorado was equipped with a very fine looking aluminum or magnesium wheel some years ago and many of these wheels are still in use and appear to be satisfactory. This was a combination wheel with the back half





Above *right*—Even the surfaces not normally seen are carefully turned on a lathe for correct balance; outer diameter is finished on precision mandrel. *Right*—Magnesium wheel, less bolt holes.

***Top*—American Racing Equipment 12-spoked front wheel ready to go dragging. *Middle*—Special cap lug nuts with long reach permit use of regular length lugs with thick mag wheel flange. *Lower*—American Racing Equipment's offering to Formula Jr.**

of the rim structure being made of steel and riveted to the front component of the wheel which was made of aluminum or magnesium.

It is interesting to trace the development and trends in this field to date and to forecast some new developments which are likely to come in the future. The Europeans were certainly among the first to use magnesium for competition wheels. The early versions, however, never gained a broad acceptance in Europe due to the fact that most of their racing equipment was and still is designed for knock-off hubs which were not adaptable to a magnesium wheel as easily as to the wire wheels which were broadly used and readily available in Europe. Most recently the Europeans have shown a definite trend away from wire wheels in search of better roundness characteristics and weight reduction. They have also shown a trend towards a more extensive use of bolt-on hubs in preference to the knock-off hub which results in considerable weight saving and reduction in cost of the car and its wheel equipment. The greatest use of magnesium wheels in Europe at the present time is in the lightweight machines under 180 cubic inches and principally for sports car racing up through the Formula I class. Dunlop has introduced a very fine aluminum wheel which closely resembles a stock American automotive pressed steel wheel in cross section in that it is a riveted two-piece drop center type. One piece comprises the rim structure and the second piece comprises the diaphragm or center section which is riveted to the rim component permanently. There are, of course, associated components for this wheel for adapting it to knock-off hubs if desired. This is a very light weight wheel; however, the price is extremely high and it apparently is not at present designed for mass marketing at a price competitive with magnesium wheels. It has been seen in use in this country on the Lister Corvettes and Jaguars in the sports car racing field in the 16-inch diameter by 6-inch rim width. On this wheel the air transfer for brake cooling must pass through a series of small round holes in the diaphragm or center section. This does not necessarily add esthetic beauty to the wheel, which might account for the lack of interest or attention to the wheel by prospective buyers in this country.

The real acceptance and use of magnesium wheels in this country can undoubtedly be traced first to the Indianapolis Speedway and to Bonneville where magnesium wheels have become virtually the standard. The early drag racers attempted to adapt wheels of the Bonneville and Indy sizes; however, the available slicks for drag racing were not compatible with these widths and a new series of wheel sizes were made.

The popular sizes at present for slicks vary from 7, 8½ and 10 inches wide x 15 and 10 inches wide x 16 inches. For front wheels of dragsters only 3½ x 15 inches low silhouette type mag wheels are popular for direct mounting to front wheel spindles without brakes.

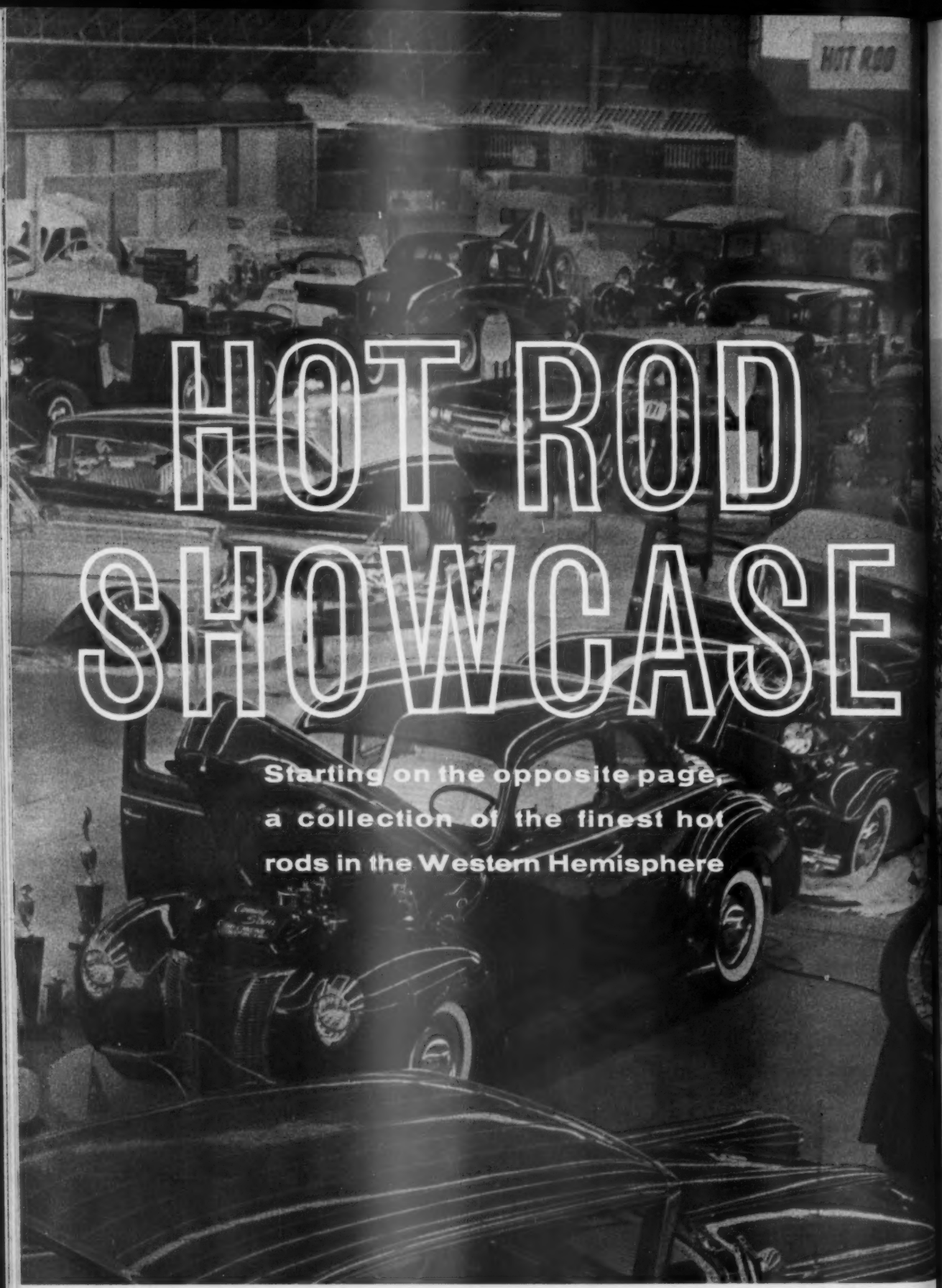
The dimensions and configurations of the rear wheels for slicks have shown an interesting evolution. The early version of magnesium wheels for slicks was 8 inches wide x 15 inches, having a solid diaphragm center section similar in appearance to a Bonneville or early Indy wheel. The wheel was arranged with a flange offset so that it could be reversed for changing rear tread width.

There developed a demand by the top competition cars for spoke openings to exchange air and frankly for a better appearance since more attention was paid to chassis design, and tread widths were not being experimented with at the strip but rather were designed to known dimensions which

were not altered. This eliminated the demand for a reversible wheel and made it possible to substantially improve the strength as well as the appearance of the wheel and to greatly enlarge the spoke openings for a maximum exchange of brake cooling air. Larger brake drums were used on drag machines which dictated the need for about 14 inches diameter brake drum clearance with the drop center offset away from the chassis to eliminate any interference between the drop center section and the brake drum. Rear tread widths were narrowed and the clearance between the chassis of the car and the inside of the tire was held to a minimum which dictated a wheel design extended off center away from the chassis. This has resulted in standards on magnesium wheel dimensions at present in the range of 3½ inches from the face of the inside bead flange to the face of the center flange where the brake drum mounts. The balance of the wheel is extended away from the chassis.

Many improvments are being offered in the magnesium wheel field. One important one has to do with the mounting procedures. Magnesium wheels are of necessity thick at the flange which introduces the problem of stock length lug studs being too short when used with magnesium wheels. One manufacturer, however, has introduced a special cap type lug nut per included illustration which makes it possible to use magnesium wheels with stock length lug studs and at the same time improve the rigidity of the mounting and appearance of the car. Another important problem has been solved with this special cap type lug nut and that is the incorporation of a heavy machined washer which is snap locked in place so that it will not fall off but yet it is free to float and will, therefore, prevent scoring of the mag wheel during mounting. Any magnesium wheel can be changed to use these cap type lug nuts by simply enlarging the lug bolt holes to 11/16 inch diameter and spot facing the front face to receive the machined washer. The cap type lug nuts are available in plain steel finish or in polished chrome finish. Because of the great variety of street machinery at the drag strips, magnesium wheels are now being used at the drag strip on many classes of cars and not just the top speed dragsters. They first appeared on modified and altered, and this developed a market for a combination wheel for street and competition. This was largely in the 6 x 15 inch series and it is gaining considerable popularity on the street at the present time. The most popular street adaptation has been on Corvettes and Ford Galaxie and Thunderbird cars although these wheels are showing up on a variety of 1960 and 1961 models throughout the country. They still appear to be a luxury item; however, it would seem that the market exists for a more competitively priced wheel in this field and it would not come as a complete surprise if something were introduced for the street machine in the next year or so which will provide roundness and light weight together with good heat dissipation and a real competition appearance at a price within the reach of the average motorist. If this trend continues for a few years it is very possible that Detroit cars will be available with this type of magnesium wheel as an optional accessory. ■

Jack Smith's Corvette, from San Francisco, sports special knock-off installation, enhancing appearance and improving the ride.

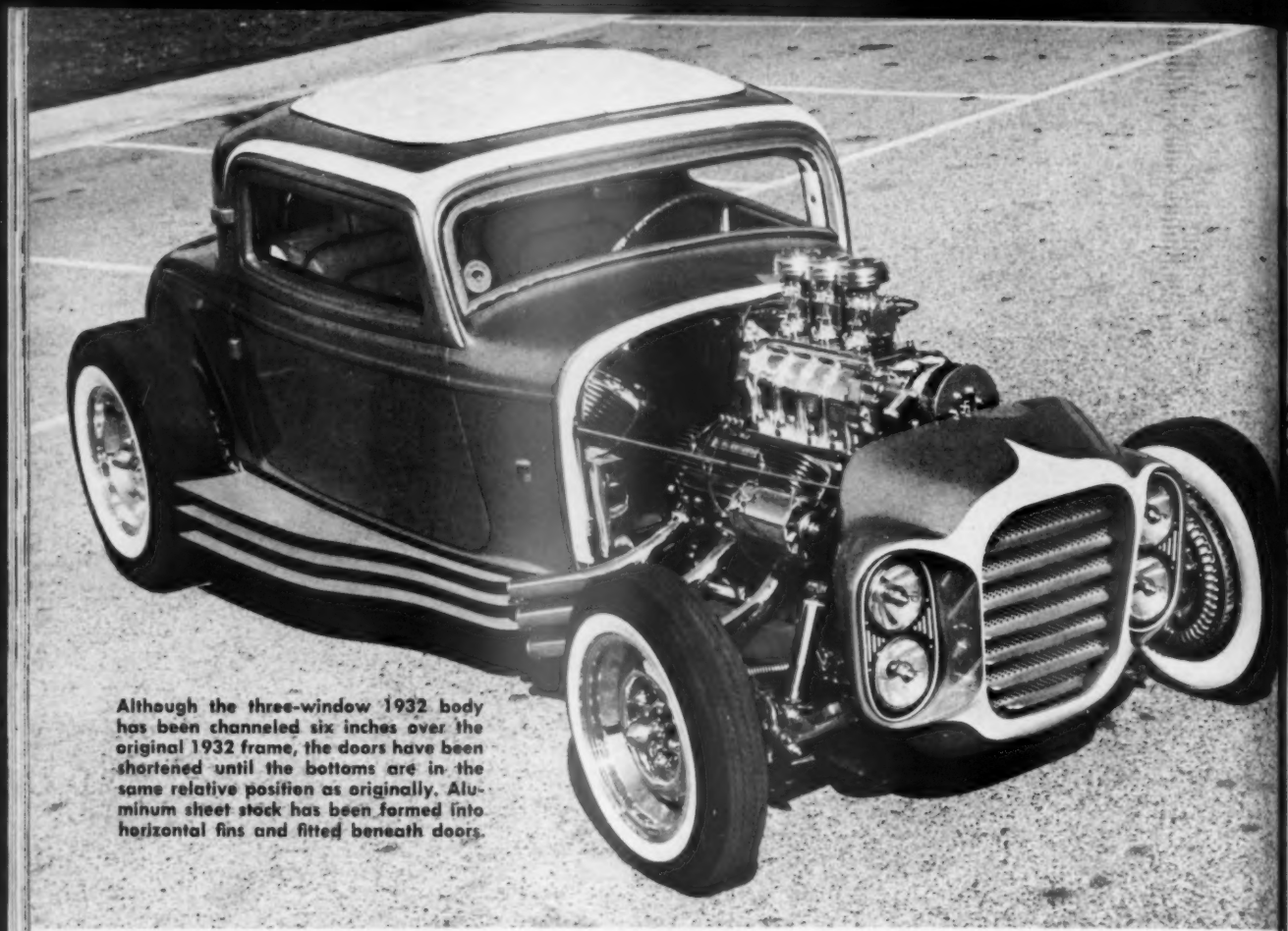


HOT ROD SHOWCASE

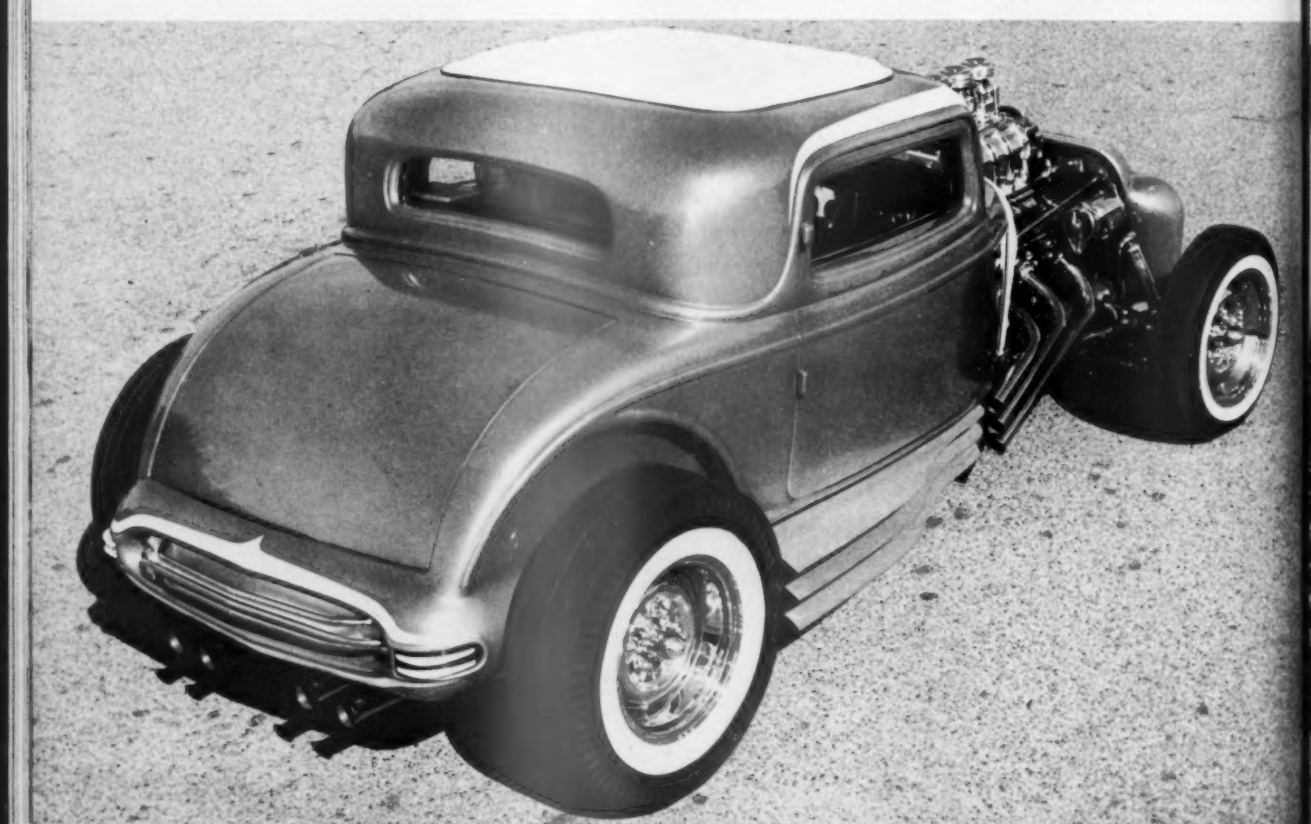
Starting on the opposite page, a collection of the finest hot rods in the Western Hemisphere

RADICAL SHOW ROD

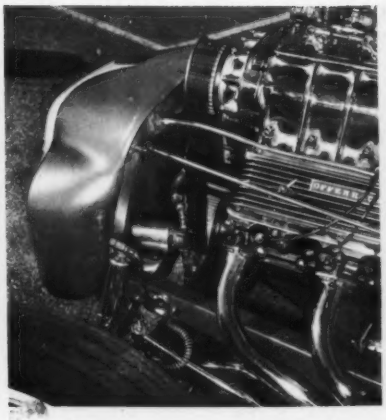
Clarence "Chili" Catallo hails from Dearborn, Michigan, became interested in rods as a natural progression of a general interest in cars. He originally purchased the coupe for $75 in 1956 when he was but 16 years old. By '57 he had built it up for street use and in '58 he was using it on the drag strip. First shows came in 1959 and now account for many of the trophies in Catallo's home. Clarence says his main problem is never being satisfied which results in constant rebuilding.



Although the three-window 1932 body has been channeled six inches over the original 1932 frame, the doors have been shortened until the bottoms are in the same relative position as originally. Aluminum sheet stock has been formed into horizontal fins and fitted beneath doors.



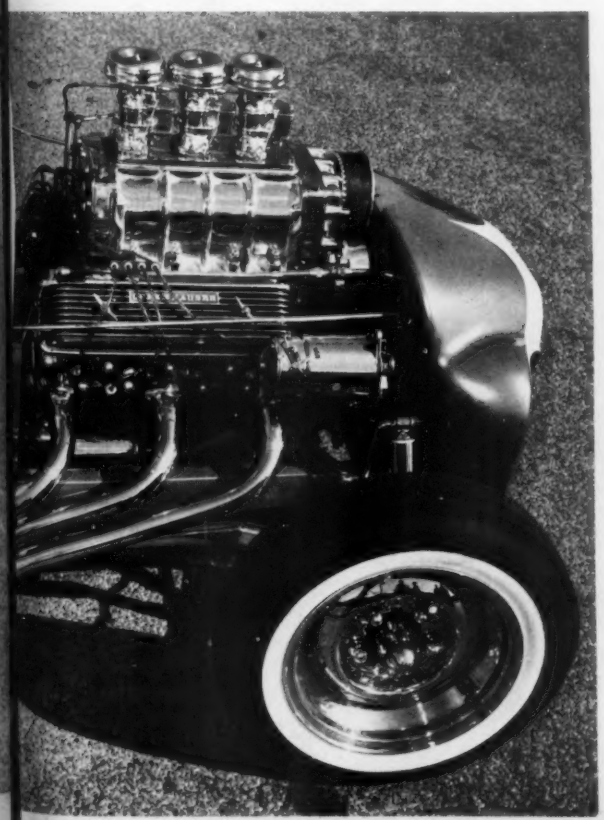
Special rolled rear pan is foundation for plastic taillight lenses and plastic base for horizontal tubing. Two mufflers make provisions for a quartet of tail pipes.



The special blower drive does not leave room for the stock Oldsmobile water pump, therefore a Jabsco electric pump is mounted on the car's front crossmember.

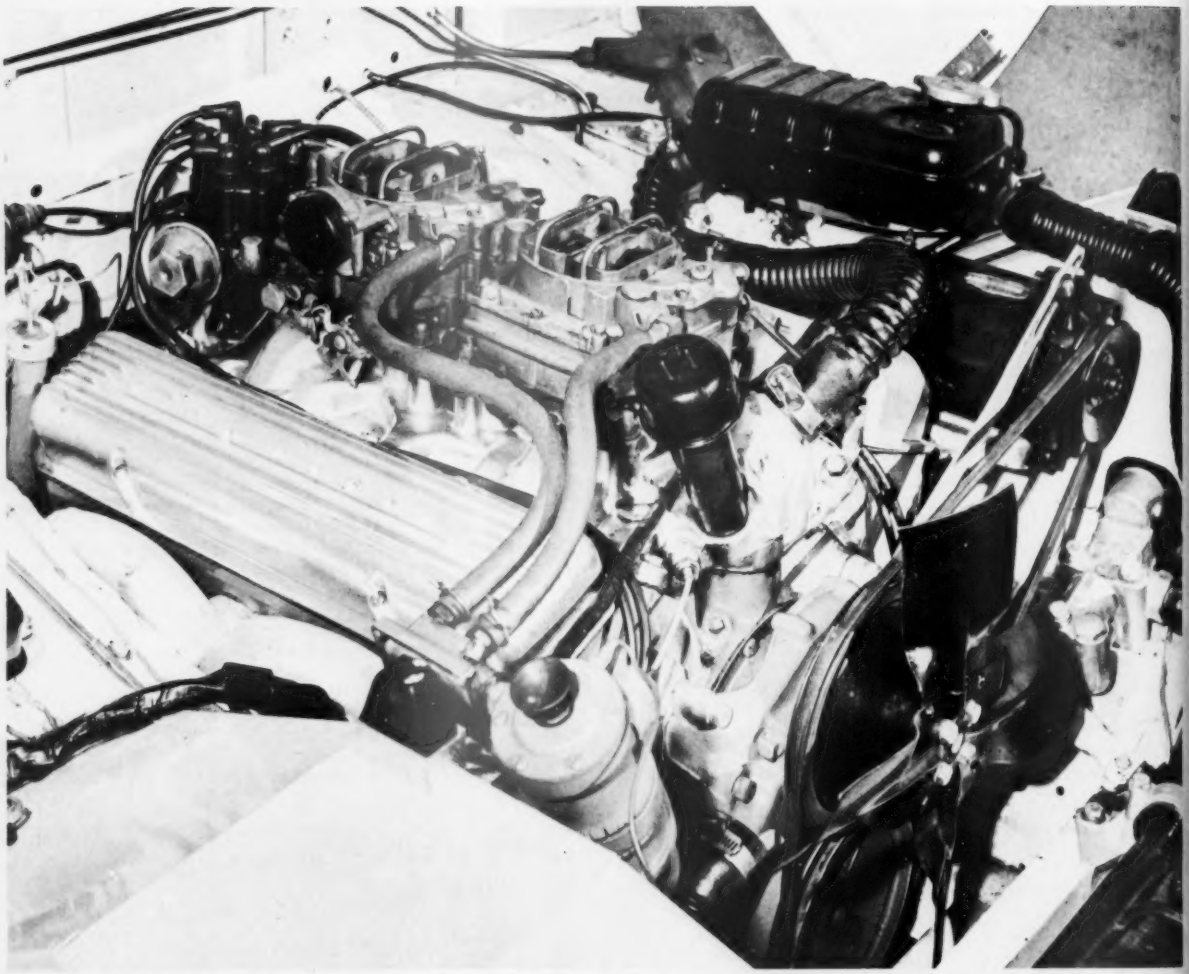
Tubular front axle is from 1937 Ford, carries '37 Ford spindles and out-of-production Kinmont disc brakes. Front wheels are Imperial centers on reversed Merc rims.

Powerplant is a 1956 Oldsmobile V8 of 344 cubic in. Chrysler intake and GMC exhaust valves are actuated by a Clay Smith cam and Iskenderian cam kit. Jahns supplied the blower type pistons with grant rings, while the 6-71 blower and installation kit were provided by Cragar. Ignition is by stock Olds.

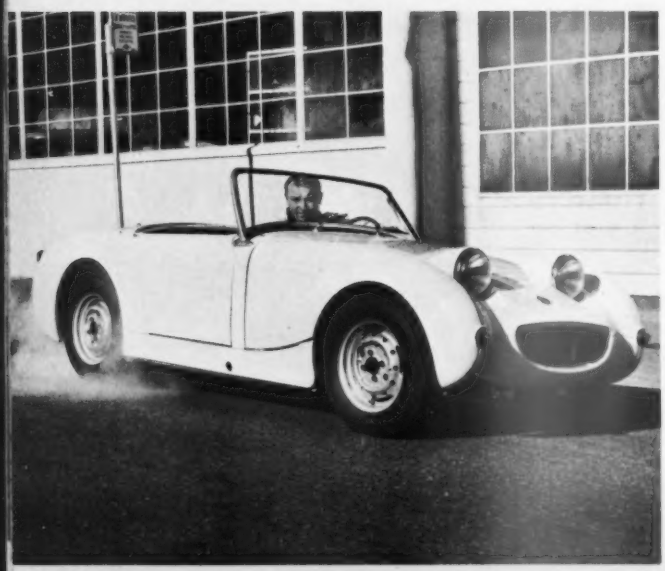


"Chili" has plenty of room in cabin that has been lowered a total of 15 inches. Interior was stitched up by Larsen upholstery shop in white Naugahyde. The stock dashboard has additional instruments plus more gauges and switches in a panel above the windshield. A 1937 LaSalle provided the car's transmission.

SPRITE CHEVY



Bob Knapp's bomb is a sleeper, even to the tone of muted Chev, until he stands on it, then it splatters rubber all over the ground! The stock BMC driveshaft and rear end were retained and now, after 1300 stormy miles, the drive train is beginning to loosen up to the extent that a huskier drive train is in order. Car's exterior is totally deceptive in appearance.

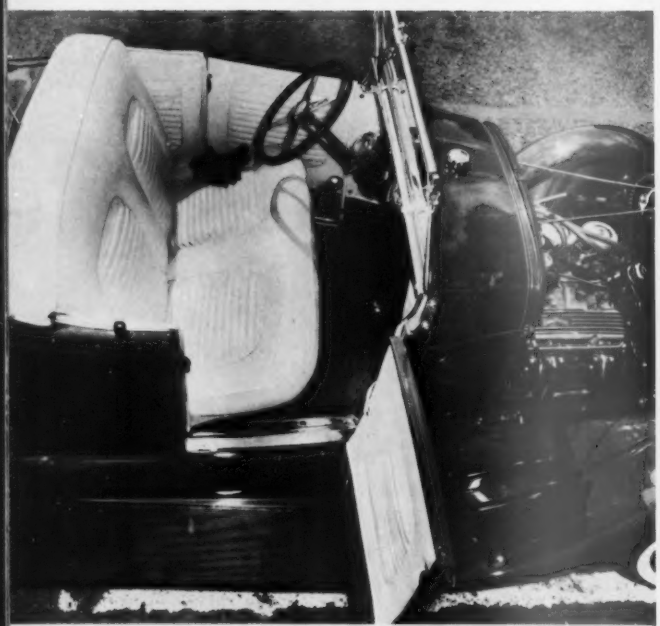


Here's that big new radiator for the Corvette mill. Surprisingly enough, once the V8 is nested in place, it looks as if it belongs. Electric fuel pump is perched on the right inner fender and the battery removed to behind passenger's seat.

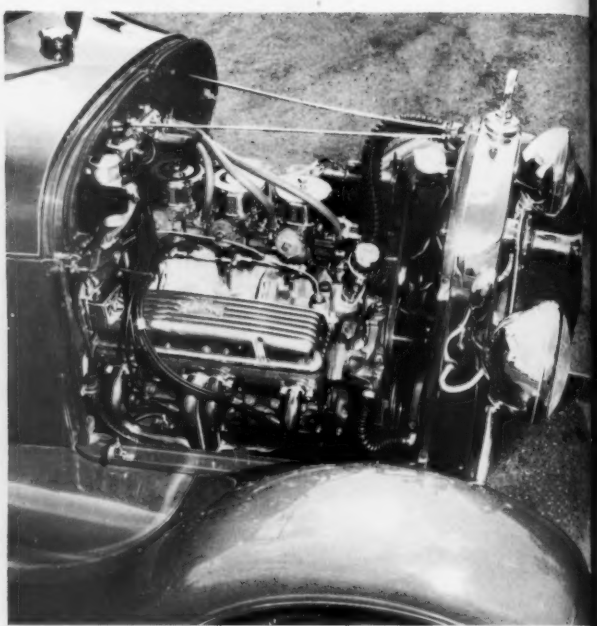
Fat Powerglide crowded the cockpit so that accelerator pedal was done away with entirely and accelerator linkage was hooked up to suspended brake. Suspended clutch was made into present brake pedal, eliminating one of the foot pedals.

PHOTOS BY CHARLES W. HAMILTON

RED ROADSTER TO CROW ABOUT



In brilliant contrast with Mandarin Red body is beautifully tailored white Naugahyde upholstery accented with fresh window frame chrome and a functional looking Bell Auto steering wheel. Ford's old cowl tank and instrument pod were kept, complemented by Sun tach on mastjacket.



The engine that gave rodding a new lease on life, the 283 Chevy V8, beds down neatly in the old '29, stock inside except for Duntov stick but sporting an Edelbrock triple intake manifold topped with Rochester carbs. The roadster's custom headers and neatly loomed wires help provide for very attractive setup.

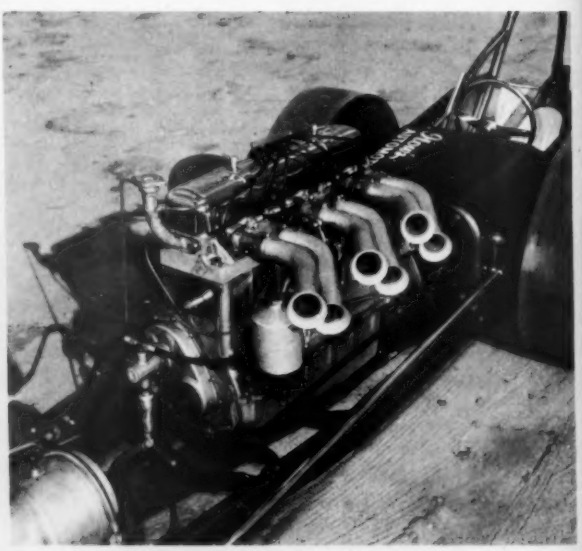
At 21 years, Paul Bos is an old timer. This is his sixth road rod. Chassis refinements list: '32 front suspension, '40 brakes, '56 pickup steering box and '40 rear end with 4.44:1 gears, all FoMoCo branded.

Born in Hawaii, Bos drew from nine years experience in hot rodding, wisely selecting '29 Ford-Chevy V8 combo as not only a good running team but shrewd investment as well. Stock body was wisely retained, enhanced by Jeep lights, Ford wheels; 15x5.90's, 8.20's.

PHOTOS BY BOB HARDEE

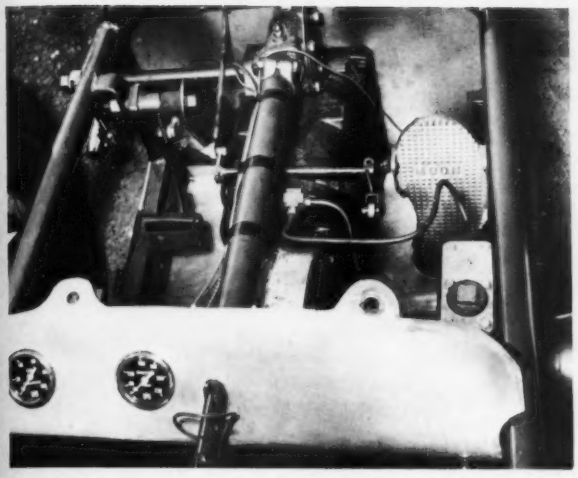
SIX-CYLINDER DRAGSTER

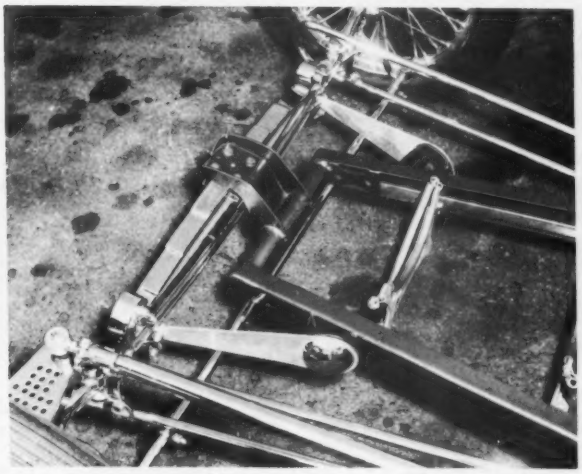
Owner Jerry Rollema works for Joe Itow's Automotive, thus the neat Itow channel frame. Wheelbase stretches 93 inches with the engine mounted parallel to the chassis and just slightly to the rear of amidships. Rear wheels are Halibrand magnesium.



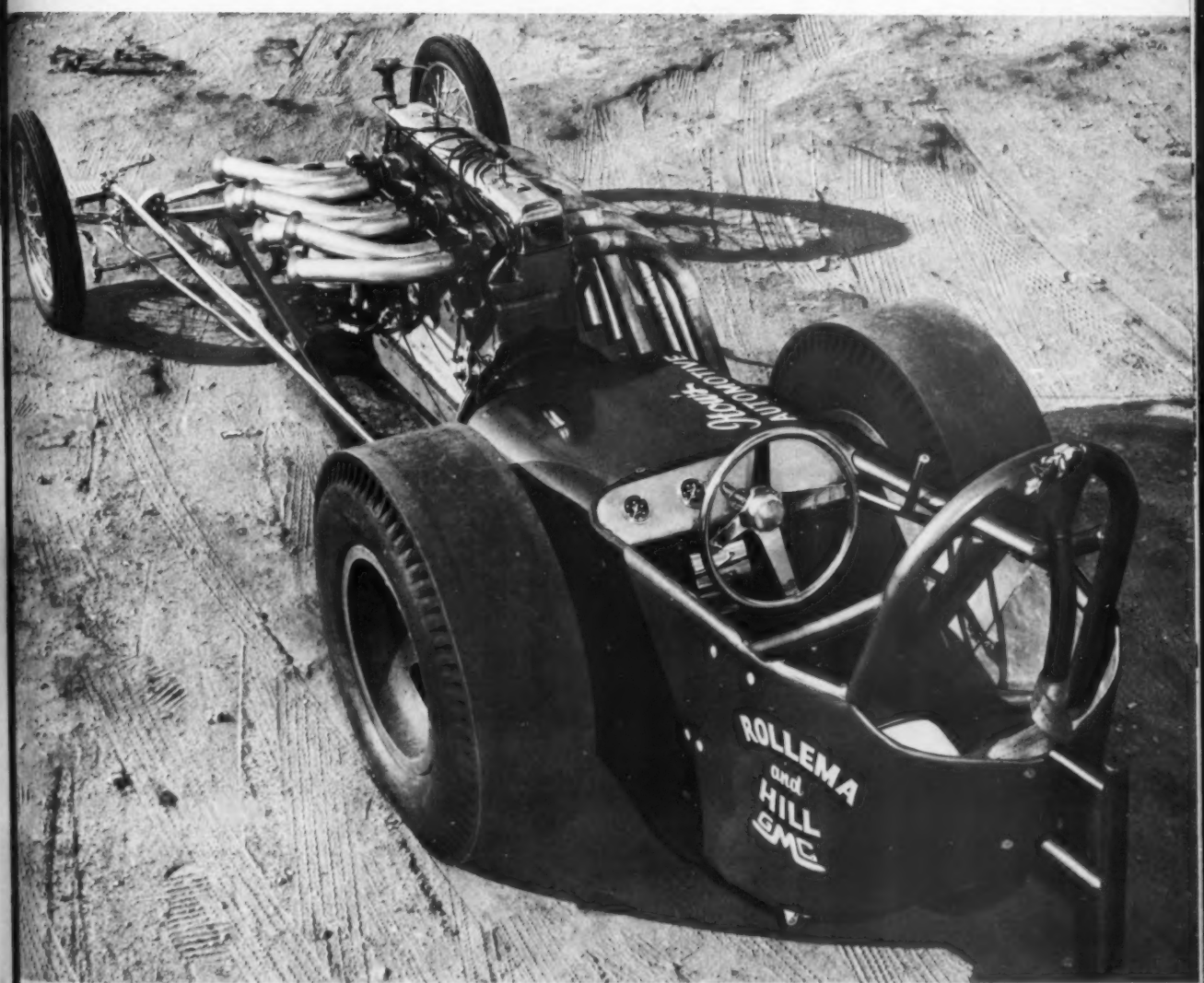
Driver Stan Hill and Jerry Rollema are seen above left. Engine of car displaces 292 inches, and runs an Iskenderian cam with stock valve train, Jahns pistons, Hilborn injectors and magneto from Joe Hunt. The twelve-port head is a product by Wayne.

PHOTOS BY ERIC RICKMAN



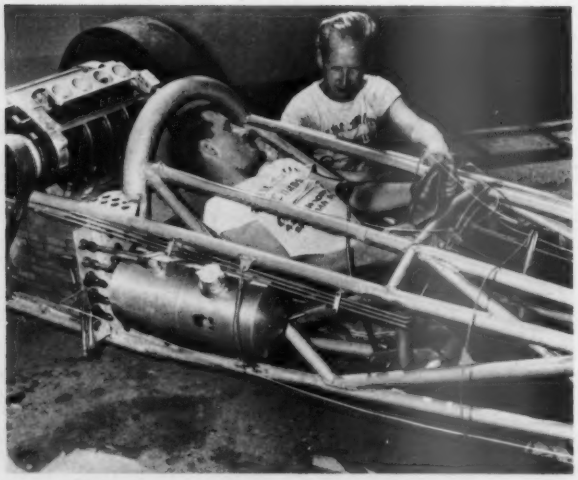


1946 Crosley steering bolts directly above the 1950 Lincoln transmission. Rollema built the trans adaptor as well as the 12-pound aluminum flywheel, purchased 10-inch clutch disc from Velvetouch. Anglia front end spreads tread 49", snubbed by friction shocks.

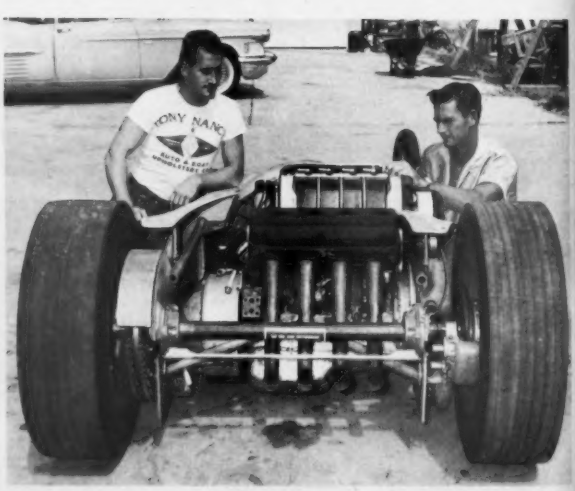


SIDEWINDER III

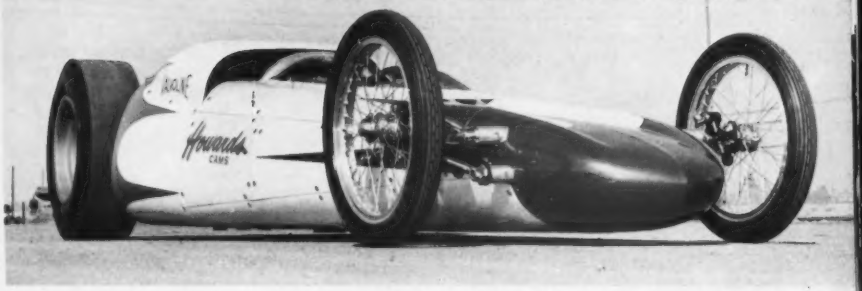
Chuck Jones tries out the reclining driving position as chassis builder Kent Fuller observes. Fuel tank is now fiberglassed as a safety feature. Fuel lines are stainless tubing and 3000 pound test hose. The brake handle works a Renault master cylinder.



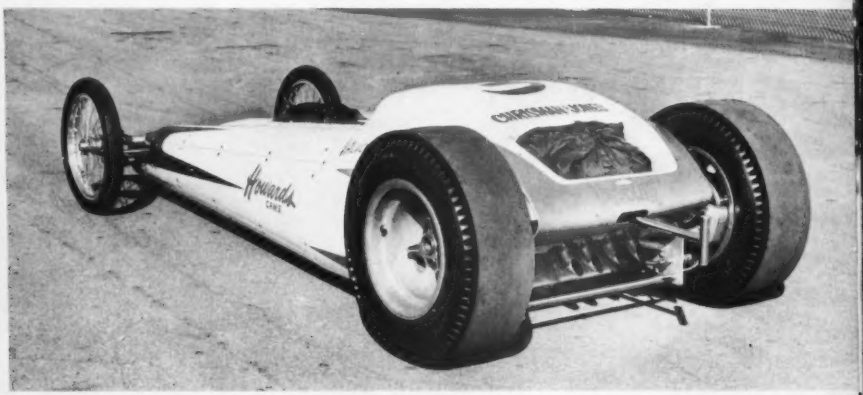
Jones and Jack Chrisman look the car over during early stages of construction. The Chrysler engine is mounted sideways, lies slightly forward to keep height down. Live rear axle is positioned by cast bearing supports fastened to tubing of chassis.



HOT ROD SHOWCASE

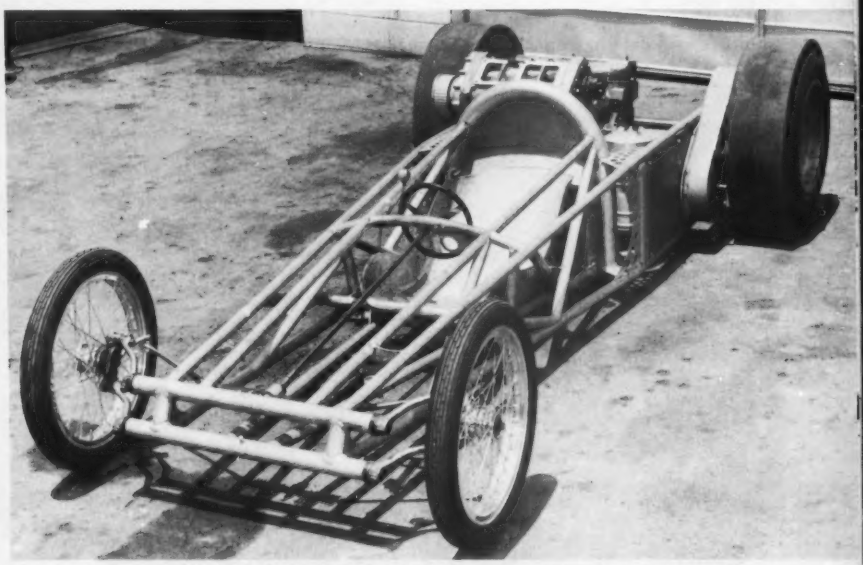


Front tires are Avon 3.00 x 19 mounted on Fuller spoke wheels laced up using special Borrani alloy rims. Full body pan skims asphalt by two inches; car is 30½ in. high.



Halibrand mag wheels with M & H slicks dominate rear. Car now uses Pontiac mill.

Merely changing sprockets will change gear ratios. Two single row chains are used, with idle roller to keep them taut.

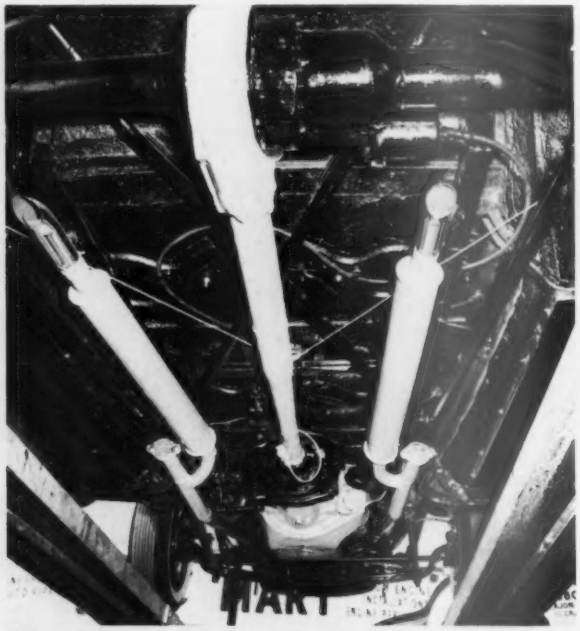


Chassis is composed of nearly 150 feet of .065 inch magnesium tubing 1½ inches in diameter. An inner cage structure surrounds the driver. Roll bar is ⅜ inch thick, 3½ inch diameter mag tubing with a ⅜ inch mag plate welded to it, forming the firewall.

PHOTOS BY ERIC RICKMAN

FATHER AND SON RODSTER

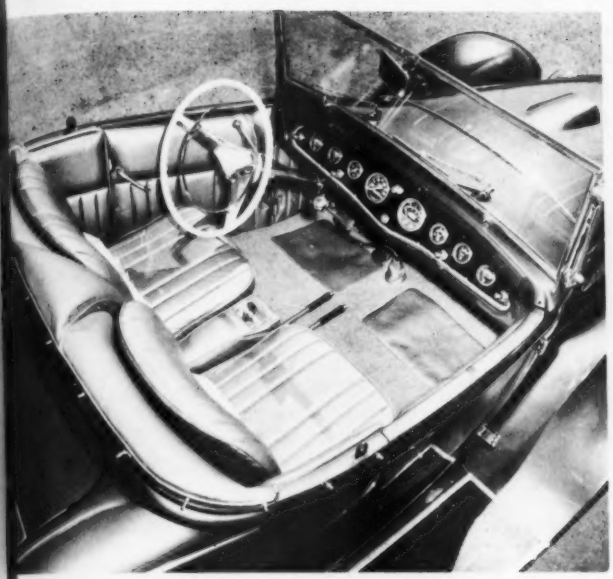
Gary DeForest's father helped him channel the 1931 Ford roadster over the rails 6 inches, insulated the union with rubber. Grille shell is a chopped and widened '32 with handmade insert. Converted fog lights provide illumination at night and parking lights pirated from a Buick lurk under fenders where they act as turn signals. Mr. DeForest constructed the hood, running boards and running board splash pans from sheet stock. Merc windshield wipers are operated by Volkswagen motor.

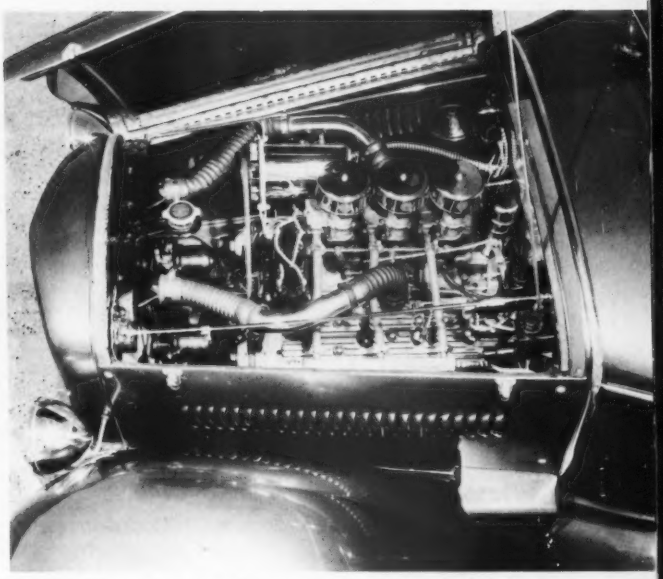


Above—Pretty Ruth Ann Spethmann is all set for cruise around the streets in one of the nicest street rods ever to come along.

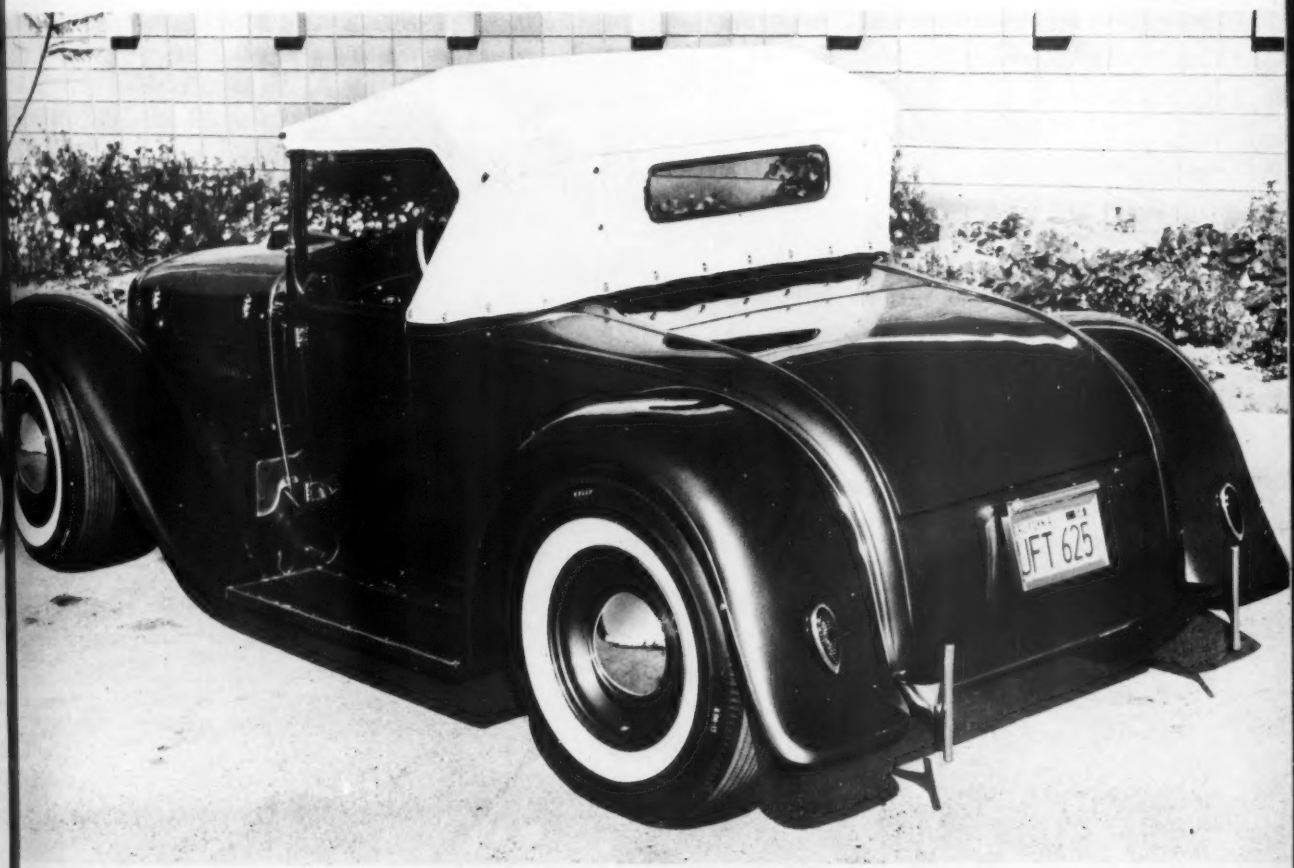
Left—1931 frame has been partially boxed, reinforced by an X member of tubing. Floor boards are from '57 Pontiac and '56 Ford hoods, properly undercoated. To get better road cruising axle ratios, a 1946 Columbia two-speed rear end has been fitted between '48 Mercury brakes. Shocks are from a 1948 Ford.

PHOTOS BY BOB HARDEE

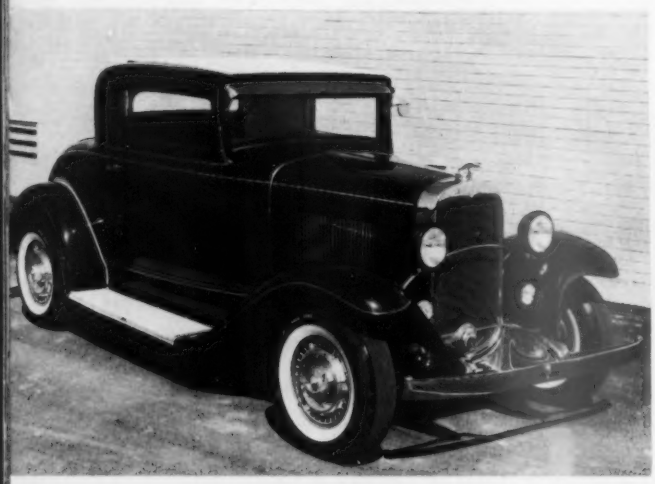


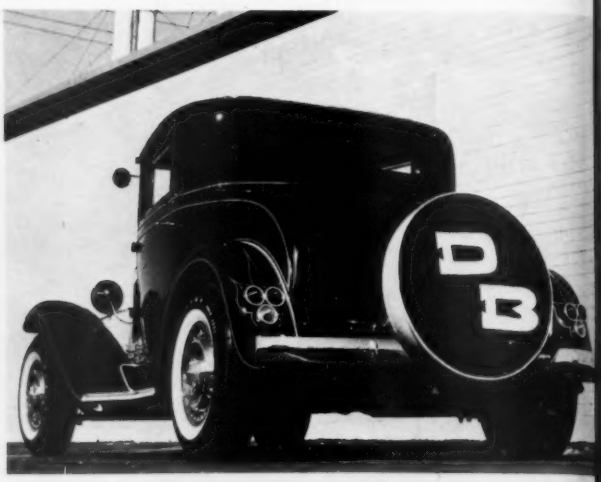


Kizer of San Diego finished the Anglia seats and rest of interior in blue-green Naugahyde. Car is powered by a '48 Merc block which holds 296 cubic inches, a Harman-Collins 408 grind cam, Sterling pistons, Offenhauser heads, and also an Evans intake manifold.



1931 CORVETTE?

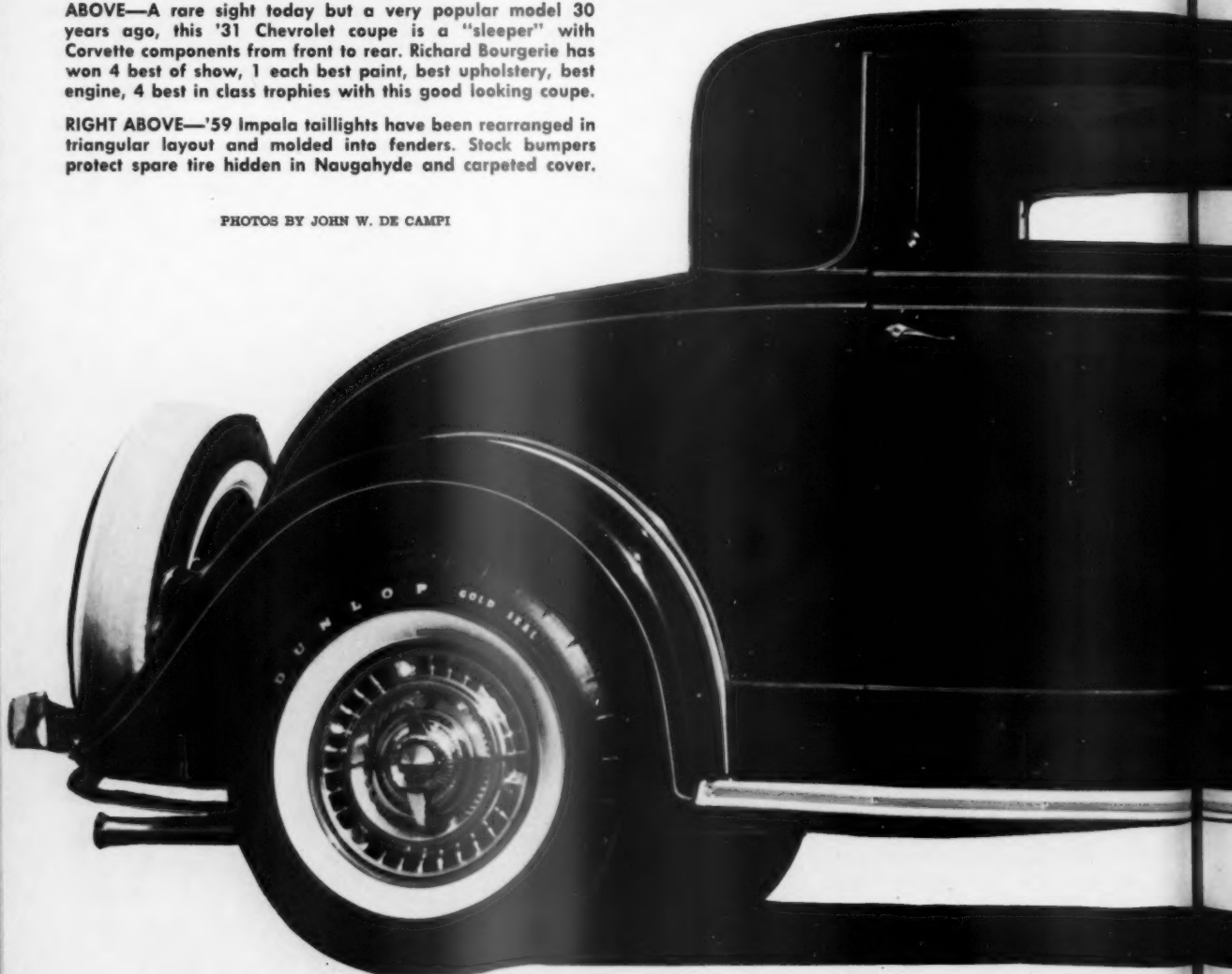


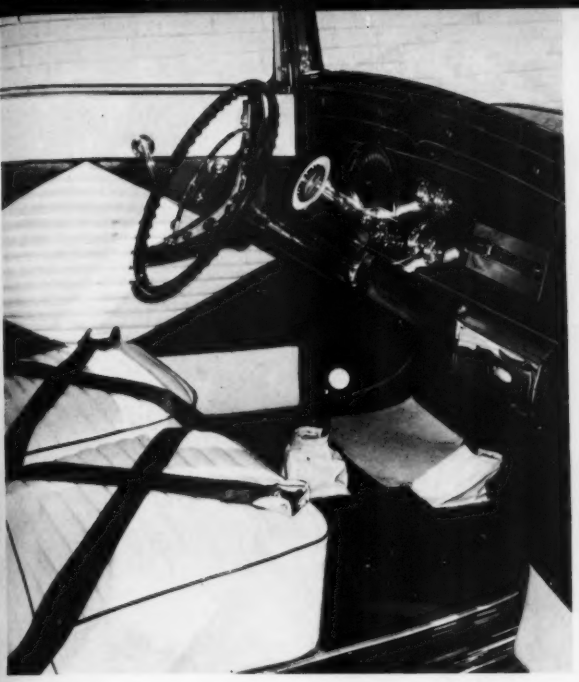


ABOVE—A rare sight today but a very popular model 30 years ago, this '31 Chevrolet coupe is a "sleeper" with Corvette components from front to rear. Richard Bourgerie has won 4 best of show, 1 each best paint, best upholstery, best engine, 4 best in class trophies with this good looking coupe.

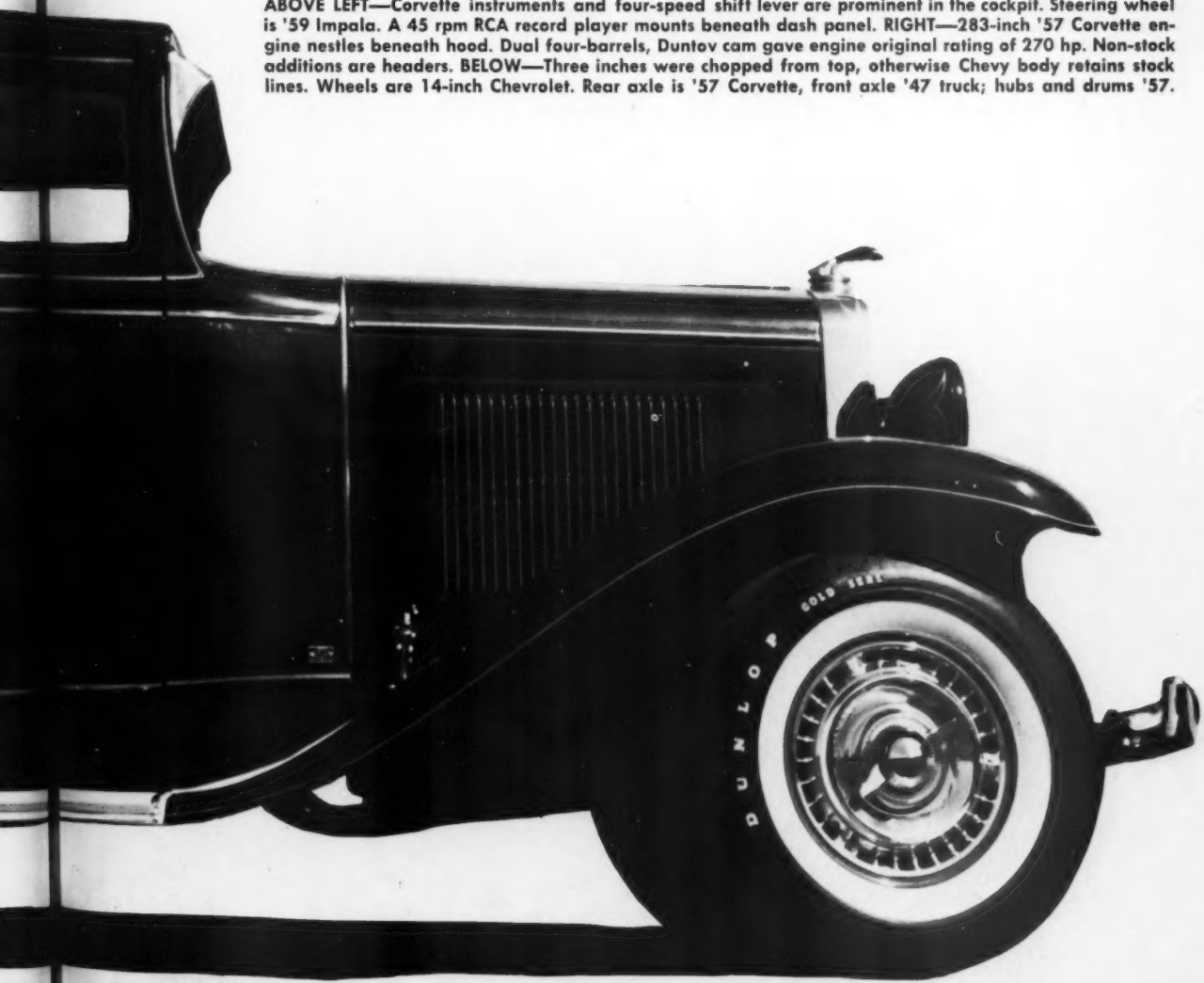
RIGHT ABOVE—'59 Impala taillights have been rearranged in triangular layout and molded into fenders. Stock bumpers protect spare tire hidden in Naugahyde and carpeted cover.

PHOTOS BY JOHN W. DE CAMPI



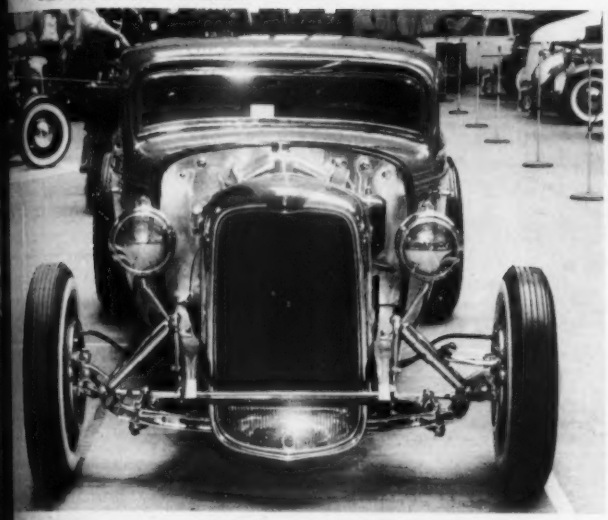


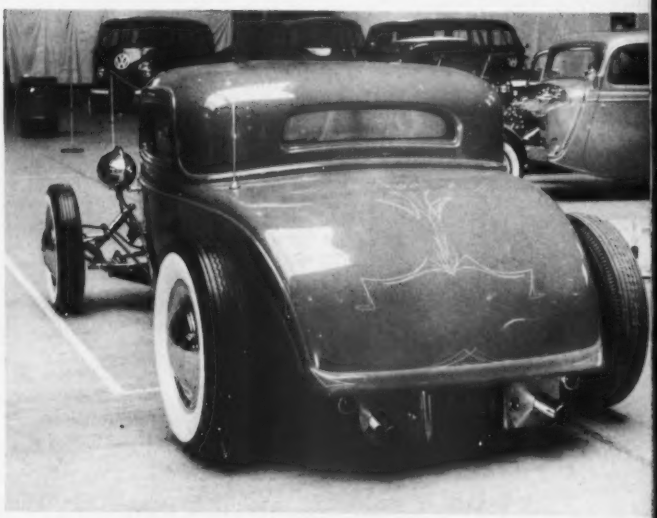
ABOVE LEFT—Corvette instruments and four-speed shift lever are prominent in the cockpit. Steering wheel is '59 Impala. A 45 rpm RCA record player mounts beneath dash panel. RIGHT—283-inch '57 Corvette engine nestles beneath hood. Dual four-barrels, Duntov cam gave engine original rating of 270 hp. Non-stock additions are headers. BELOW—Three inches were chopped from top, otherwise Chevy body retains stock lines. Wheels are 14-inch Chevrolet. Rear axle is '57 Corvette, front axle '47 truck; hubs and drums '57.



GLAM
OLD TIMER

PHOTOS BY ERIC RICKMAN



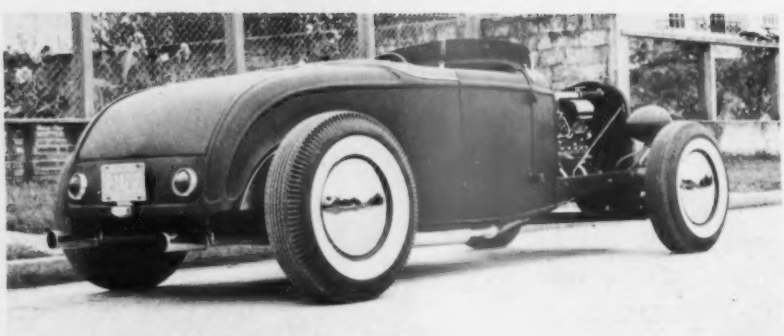


LEFT—A seasoned veteran of show circuits is Andy Kassa Jr.'s brilliant red Ford from Passaic, N. J. His deuce has taken first in such huge shows as Madison Square Garden and has hit 107 mph in quarter mile drags. ABOVE LEFT—Startling lowness of Kassa's coupe becomes most evident head-on. By keeping the flathead V8, Kassa was able to retain original wishbone with drop axle, custom steering arm. ABOVE RIGHT—A total of over ten inches was deleted from the deuce's stature by chopping the top 3½ inches and channeling the body 6¾ inches over frame which was then Z'd at rear. Note solid rear deck and special panel below. RIGHT—Blue and pink genuine leather upholstery helps cinch the deal when it comes time to pass out the trophies. BELOW—Beneath the spreading air-horns a lusty flathead lies. Immaculately detailed '48 Merc is athletic type now with big bore and stroke, Harman-Collins cam, Jahns piston and Offenhauser heads running 10:1 compression. Carrying four Stromberg 97's is a Weiand manifold. The husky impulse is passed to a Weber aluminum flywheel and stock gearbox, with Ford rear end, 6.70x16 tires.

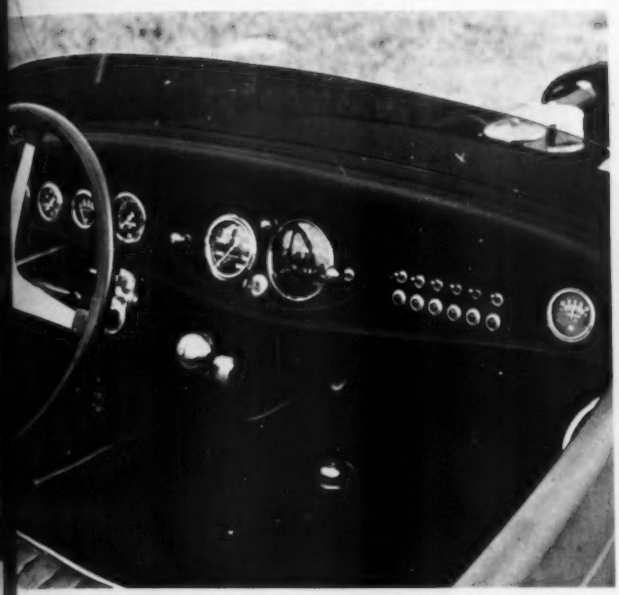
GOER FRO
GUATEMA

Captain J. F. DeLarm, Guatemala, built this '32 Ford primarily for sports car racing on rough Central American roads, by ways.

Capt. DeLarm decided to try his own form of unitized construction for maximum strength. Channeling the '32 body eight inches, he welded body and frame together as one solid unit. Front axle is dropped, rear lowered by reversing the axle housing to drop the spring hangers.

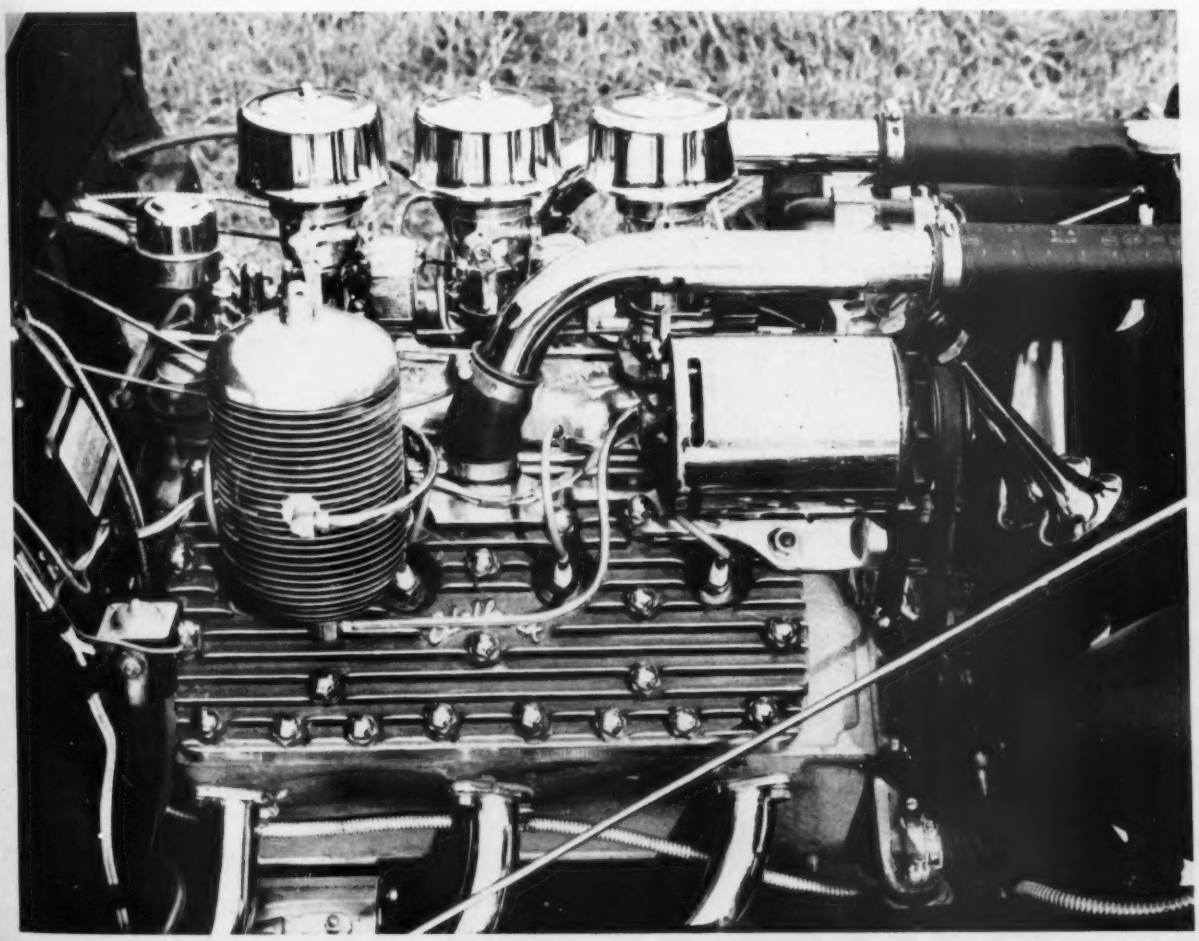


Rear deck of the deuce contains battery and aluminum oil tank from a C-47 which acts as fuel tank. Windshield is just tall enough to deflect the soft tropical breezes.

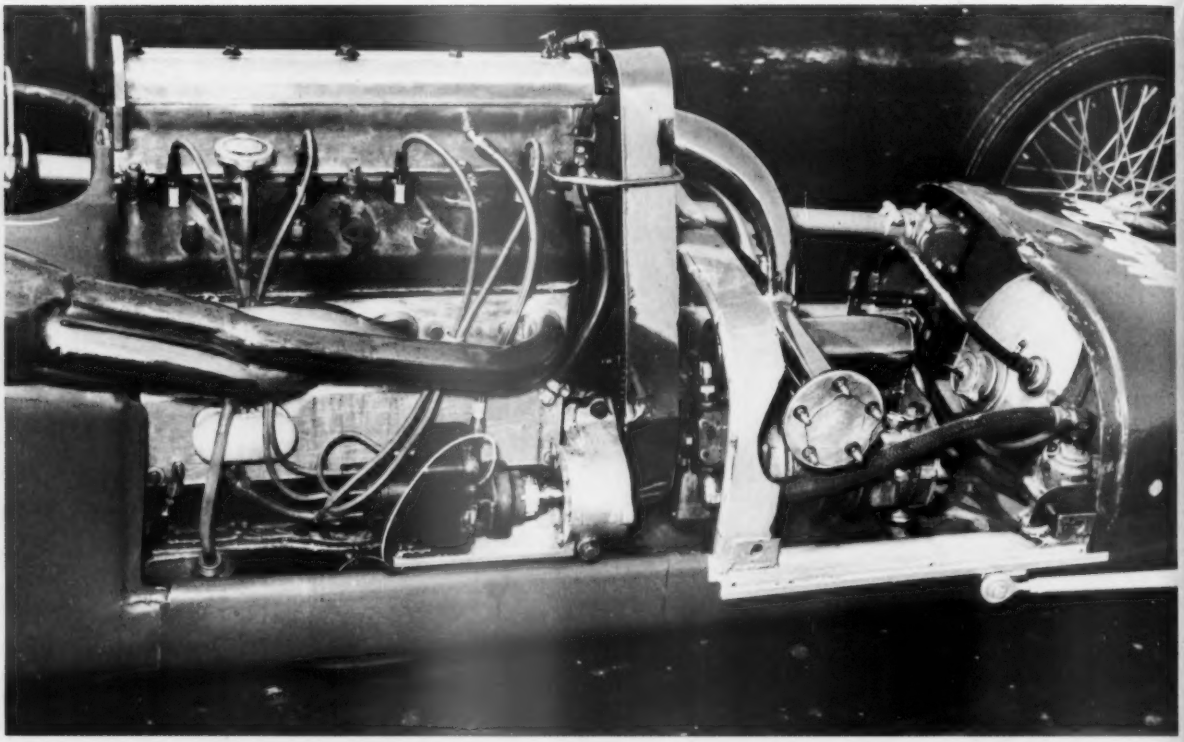


Bell Racing wheel tops the '32 steering column. S-W instruments with speedometer registering in kilometers are set into dashboard. Each light switch has indicating light for on, off.

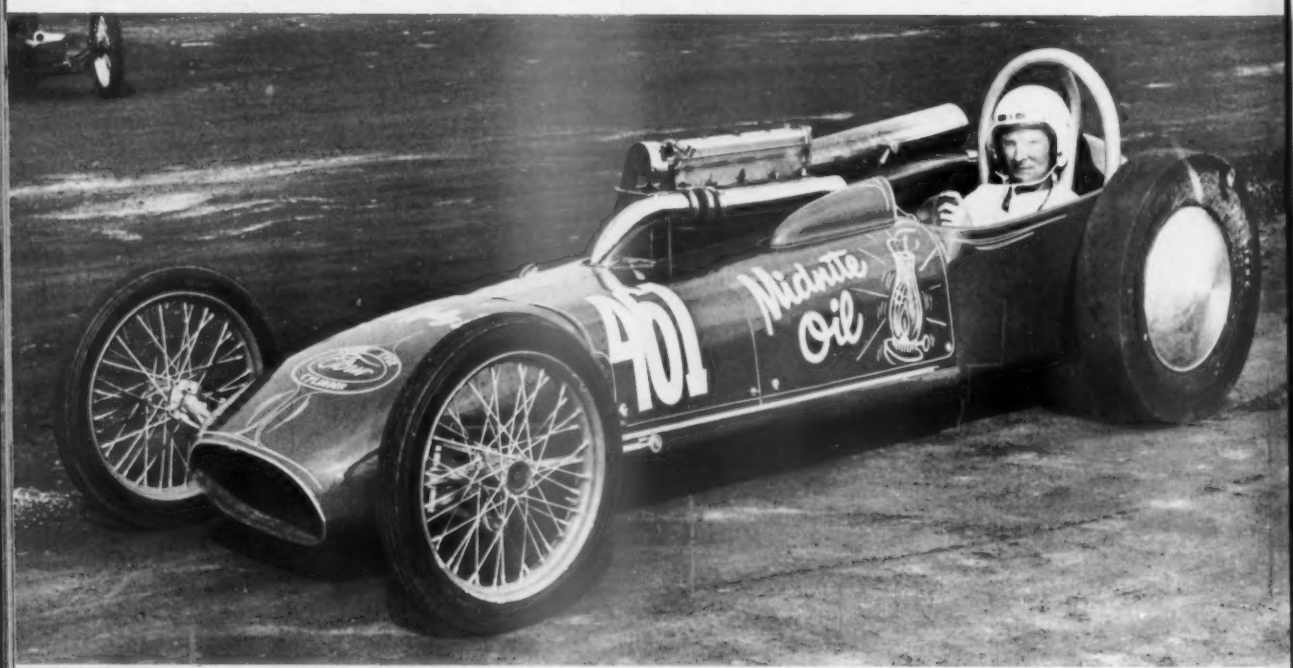
Capt. DeLarm caught the flavor of American Roadsters very well. Interior has rolled and pleated seats, panels. BELOW—3/8-th flathead has 296 cu. in. Heads, manifold, Edelbrock.



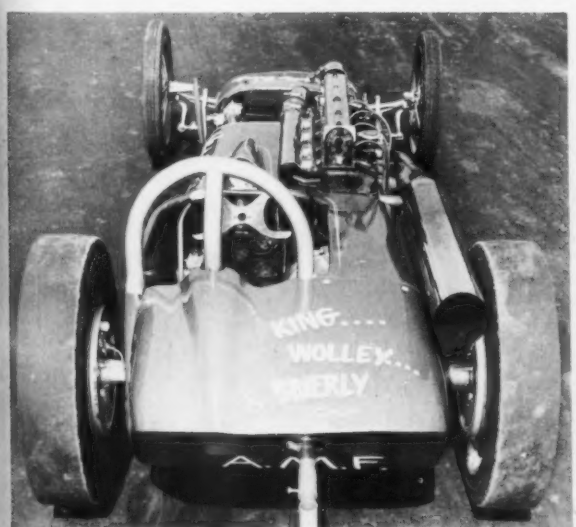
MIDNIGHT OIL



ABOVE—Heart of K-W and B stormer is a '32 Ford "Banger" of 220 cubic inches. Sparks overhead cam head is of 12.9 to 1 compression ratio, tops pistons of Jahns manufacture holding 1960 Pontiac rings. Enderle injection supplies fuel, mag is by Wico. BELOW—Victor King, handler of four banger in position. Aircraft wingspars provided frame rails, stretching the wheels 96 inches. Front axle is from English Ford mounting spoke wheels, Pirelli tires. Red enamel body panels are held in place by Dzus fasteners.



This little rail resembles an Indy car more than a dragster, but drag performance is strictly first class, having registered a speed of 107.65 at 1960 Nationals. This stood as the National record in the popular X Dragster class until the Tempests came along.

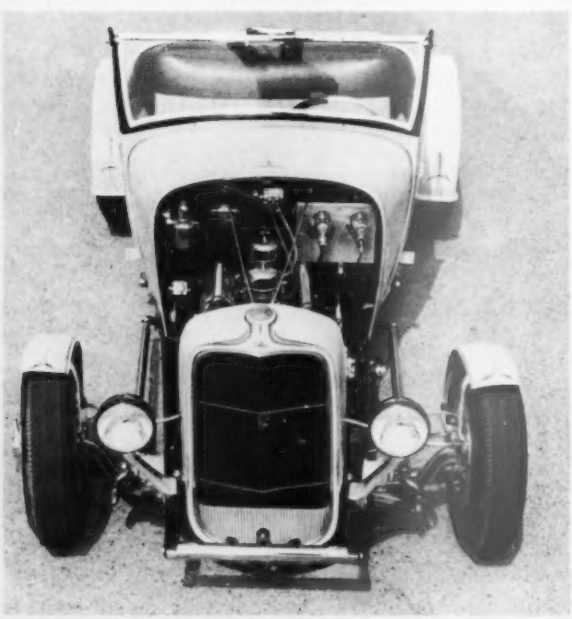


1942 Lincoln brakes provide stopping friction for slicks mounted on 15-inch Mercury wheels. Estimated horsepower is 250, makes a commanding signal through the sufficiently large exhaust pipe. Because of ancient vintage, the builders had to spend long hours constructing unavailable speed parts.

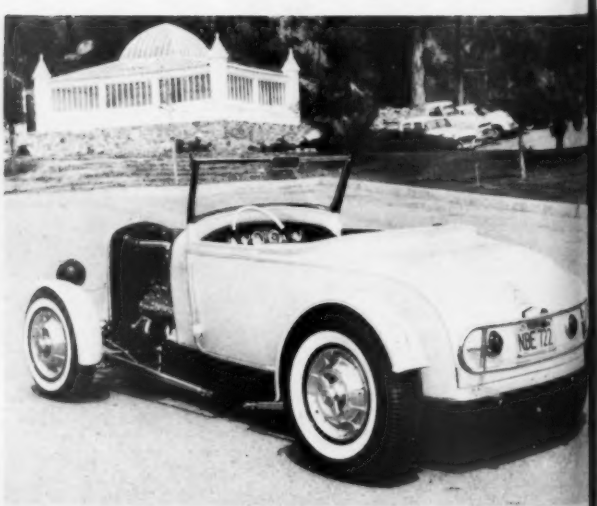
A Low quick-change fits the narrowed Model A rear end. Transmission is from a '36 Ford using 2nd and high gears only. Look closely and you'll see that the 360 degree scatter-shield extends rearward alongside the gearbox and also covers the rear end center section. Steering is from '49 Crosley.

PHOTOS BY ERIC RICKMAN

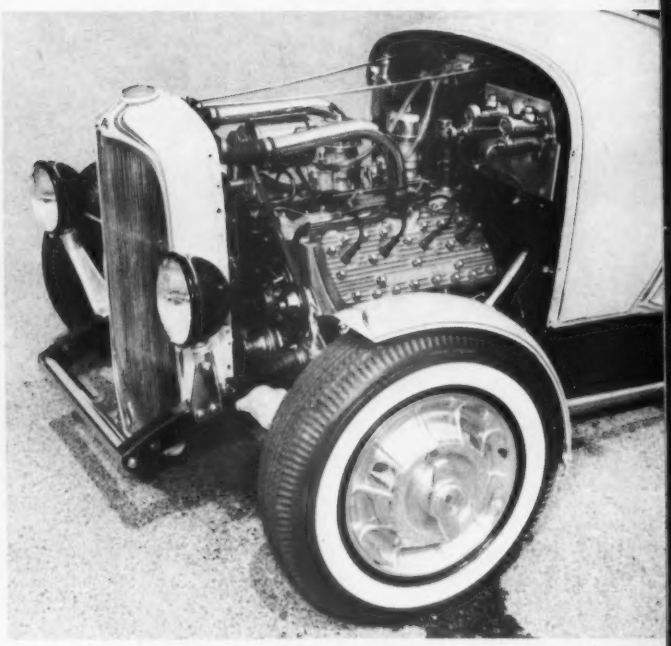
SPORTS ROADSTER



Gary Buckland and Mary DeMangus pose with bright yellow '29 roadster. Close-fitting fenders front and rear were fashioned to mount on brake backing plates and move with wheels. Cast aluminum brackets support headlights alongside '31 grille. Rear wheel wells in body have been smoothed and deck opening filled. Husky nerf bars protect rear of roadster.



PHOTOS BY ERIC RICKMAN



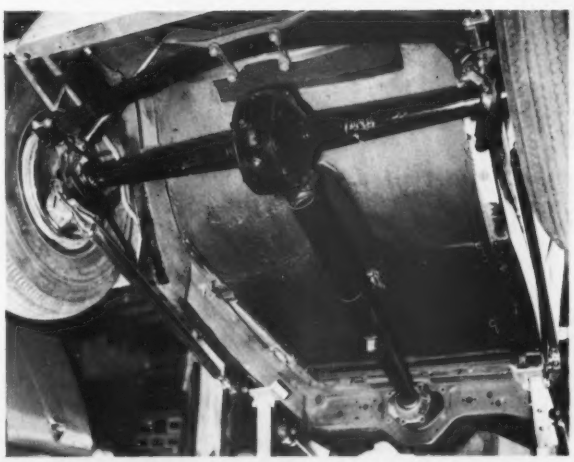
ABOVE—Bill Colgan of Burbank did the tuck and roll interior with yellow and black Naugahyde. Custom dash panel was designed and installed by Gary's dad with a full set of instruments. The floor shift sits on top of a '39 Ford trans.

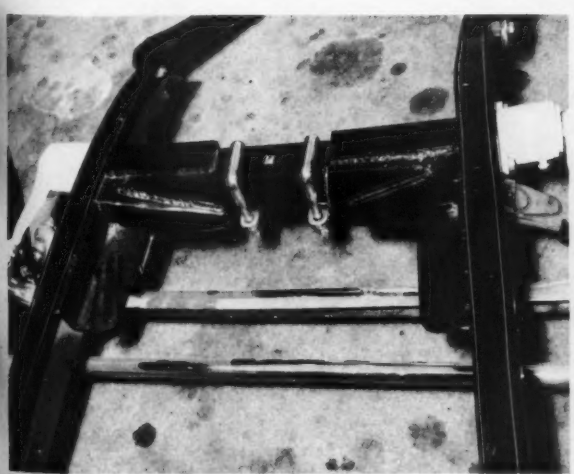
RIGHT ABOVE—The '48 Merc flathead has stock 239-inch displacement but is fitted with Novi pistons and Offenhauser heads for 10:1 compression; Fenton dual intake manifold. A thick aluminum plate fastened to the firewall provides a very sturdy mount for the brake and clutch master cylinders.

RIGHT—Modifications at rear for '40 axle include Z'd frame, plated for strength, split radius rods, relocated crossmember.

BELOW—Crossmember was welded into '32 frame for '40 spring. Flat crossmember gained extra 2½-inch drop up front.

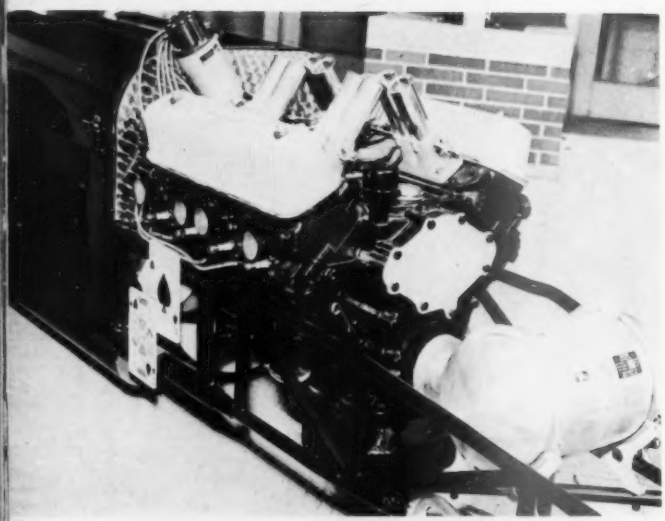
BELOW RIGHT—Swing pedals for clutch and brake were made from tubing with steel plate brackets. Three-bolt pattern on the master cylinder holds brackets solidly to firewall. Original cowl tank and instruments were trimmed out of the body.



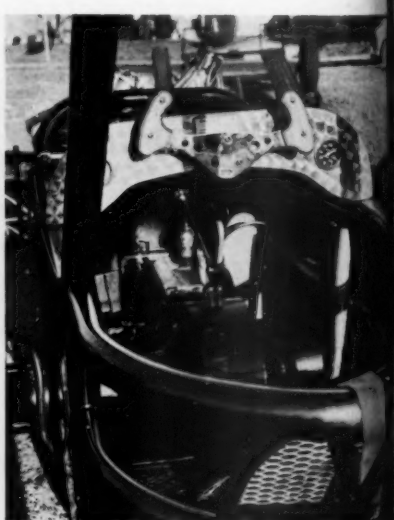


BLACK JACK

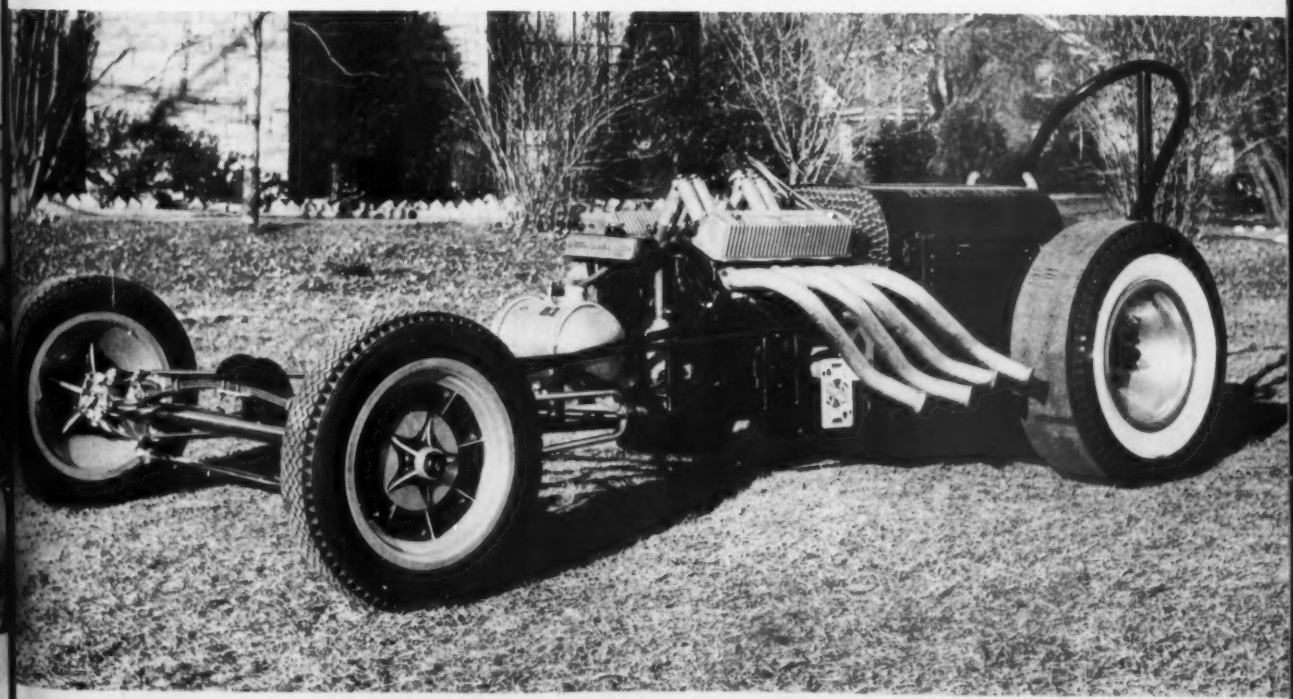
PHOTOS BY BOB HEGGE, BOB DANIELS



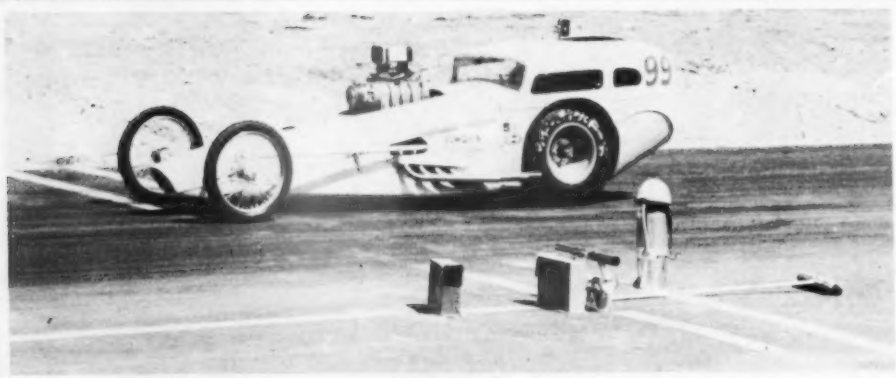
ABOVE—Photo taken during early stages of construction shows the small and rigid frame construction, heavier and very strong roll bar structure. '37 LaSalle transmission takes torque via the Schiefer flywheel clutch assembly. LEFT—Engine is a 1957 Lincoln with 368 cubic inches. A Clay Smith cam teams with Jahns pistons, Grant rings, Hilborn injection and Vertex magneto to build up steam for acceleration sprints. Note junction of small and large frame tubing near front spark plug. Gas tank is Moon. LOWER LEFT—Jack Teagardner has been car enthusiast for 20 years. Previously competed in a supercharged 1957 Merc Turnpike Cruiser C Gasser. BELOW—Engine-turned dash tops cockpit containing narrowed '46 Ford rear end and a '55 pickup steering gear modified to center-mount style. Wire mesh forms a strong seat bottom and back. RIGHT TOP—1946 Ford spindles are pinned to homemade front axle of 1¾-inch diameter tubing; tread, 54 inches. RIGHT BOTTOM—Handiwork of Jack Teagardner and Frank Held, 105-inch wheelbase dragster features frame of ⅛-inch Shelby seamless tubing. Front wheels are Halibrand mag holding 4.50 x 15 Dunlop tires, rears are Halibrand with M & H slicks.

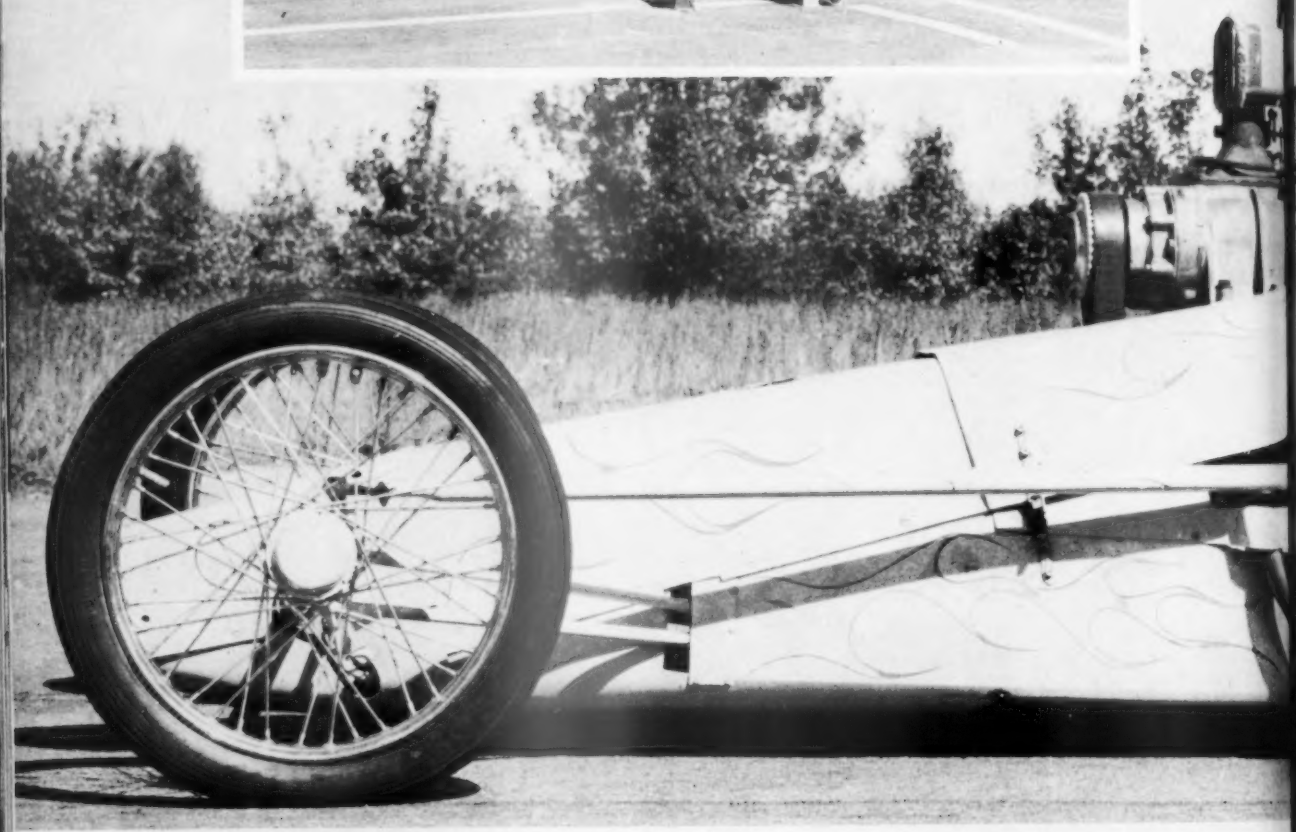


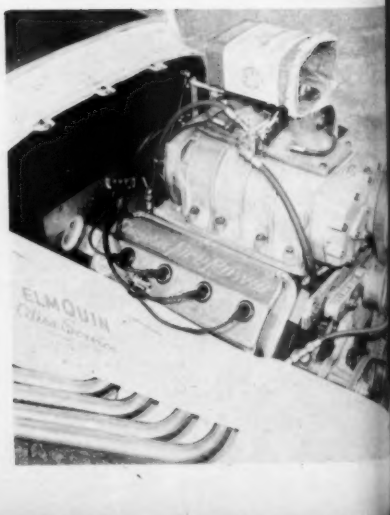
HOT ROD SHOWCASE

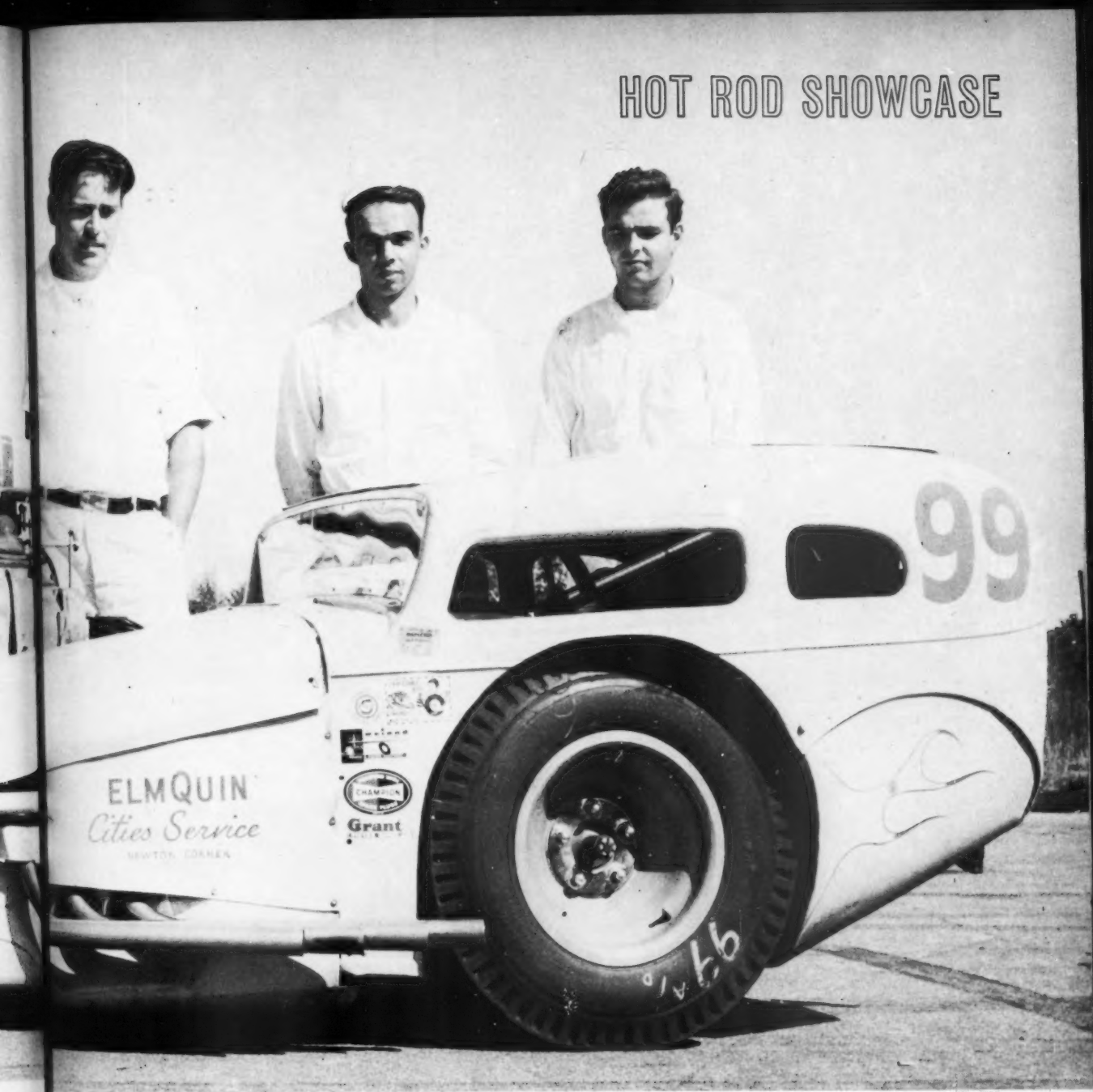


THE NINETY-NINE





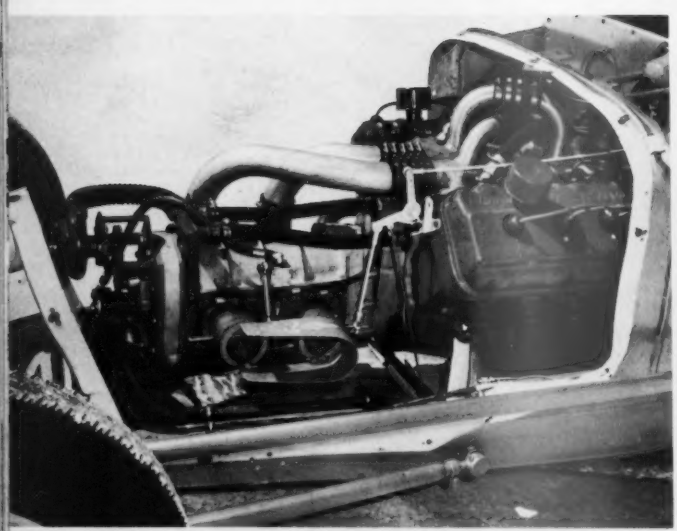




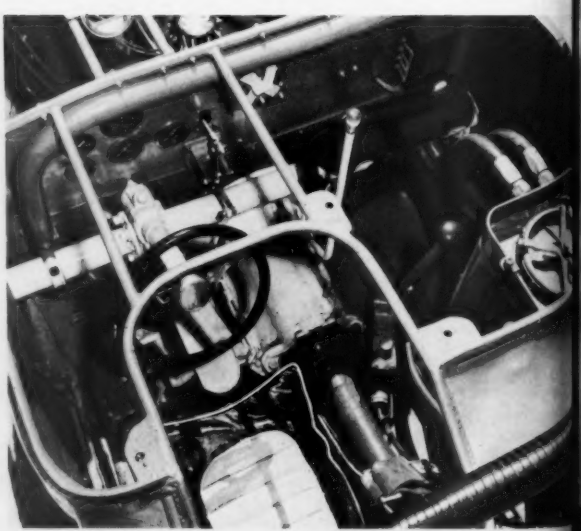
Number 99 carries a '54 Chrysler, has hit 148.90 mph in 9.92 seconds. 331 cubic inch engine uses an Iskenderian 5-cycle cam and kit, Grant rings, Weiand intake manifold, Isky blower drive and Hilborn injection. Flywheel made by Crower and a Weber 10-inch pressure plate is used. Cragar adapter holds '48 Cad transmission. Mag is Scintilla Vertex. 1930 Bantam body has been cut away radically, top chopped 8 inches with entry through top. The eastern competition coupe has $2500 invested in it now. Holes drilled in the rear allow trapped air to escape. Car owned by Ninety-Nine Racing Team, Belmont, Mass.

PHOTOS BY JIM CHASSE

WEDGE-SHAPED 'T'



Brute power is needed for all-out speed runs. This rod has it with '54 Chrysler of 398 cubic inches running a Racer Brown roller cam, Herbert rollers, Smith push rods, needle bearing rocker arms, Forgedtrue pistons, Grant rings, a front mounted blower held by Potvin kit. Stu Hilborn injector is utilized.



Office of the screaming 'T' is strictly utilitarian, with Bell wheel topping the modified '32 steering gear. Cooling water is held in can at right and fire extinguisher rests on firewall. A Schiefer flywheel and pressure plate sandwich a Velvetouch disc ahead of '39 Lincoln Zephyr transmission. Brakes are hand operated.

PHOTOS BY DON FRANCISCO

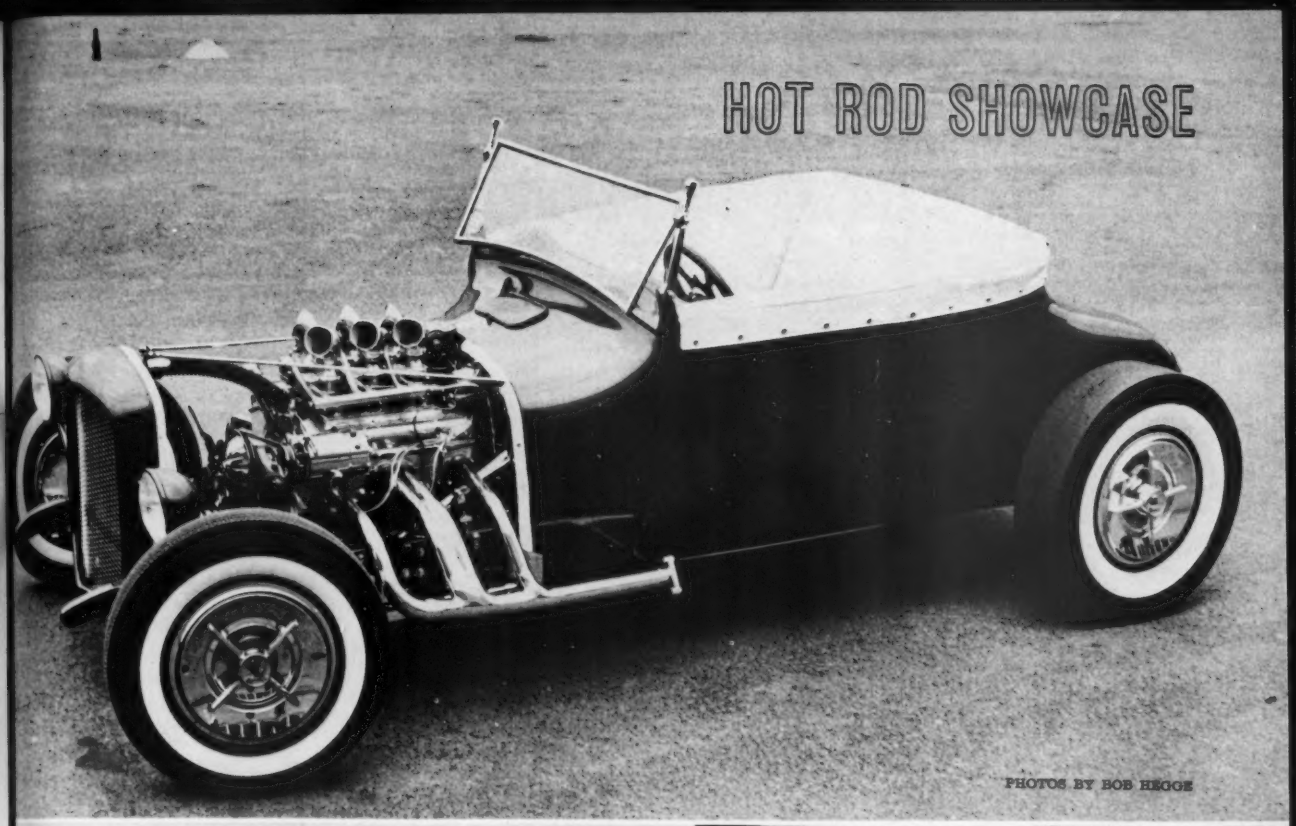
Fred Larsen and Don Cummins are old hands at smashing dry lakes and Bonneville speed records. Modified 'T' roadster hit awesome 212 mph at Bonneville in 1960. Wedge shape of machine helps split the wind nicely. Front axle is a dropped '32 wearing '40 Ford spindles. Louvered deck lid and rear panel shield a Halibrand quick-change carrying 3.27 gears. Full length belly pan helps car's handling ability at high speed. Radius rods have been split and mounted to the side rails of the 1932 Ford frame.

HALL'S S -STOPPER

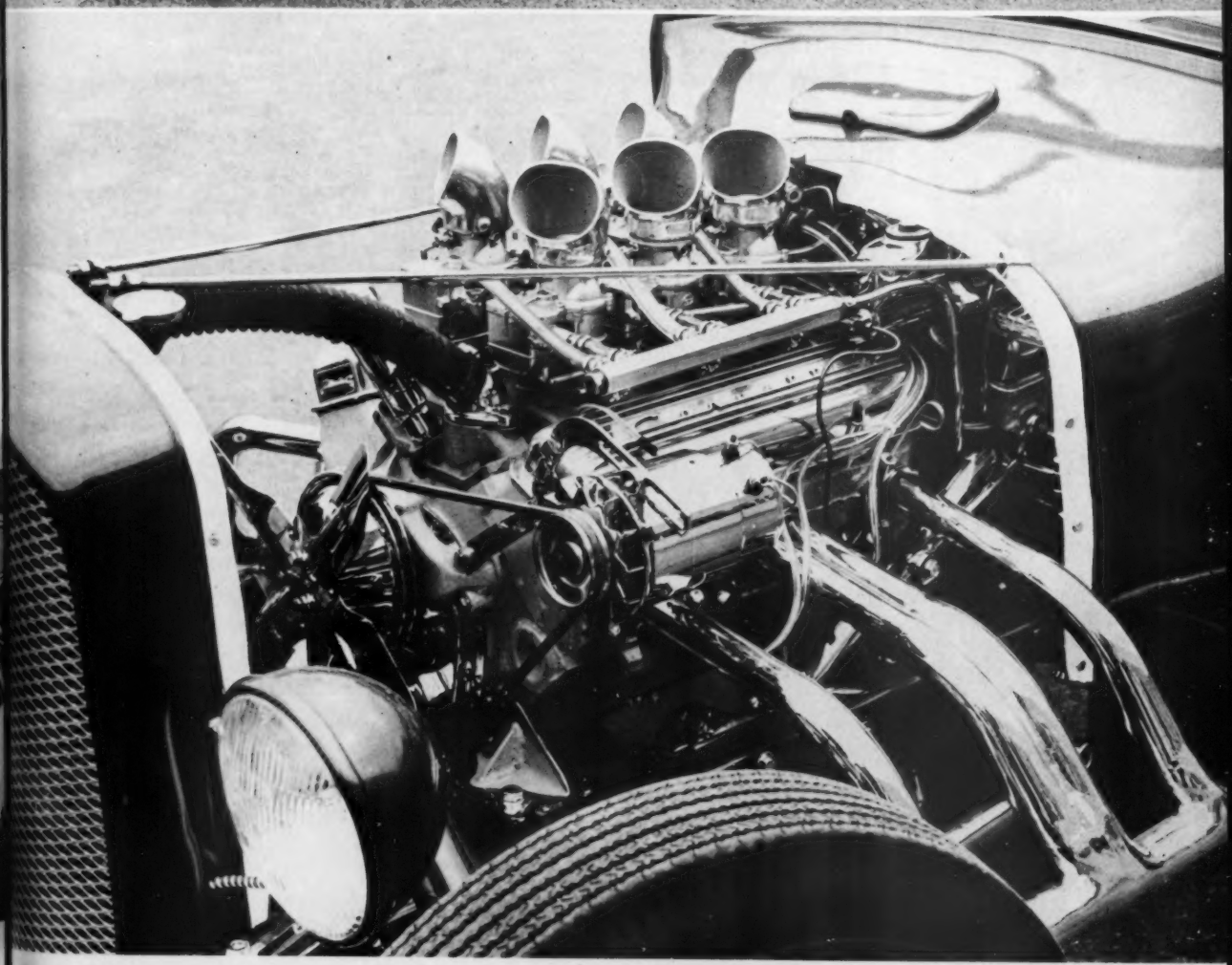
John Hall built this show-stopping '27 'T' at a cost of $1,400 after an original cost of $40. He did all of the work by himself.

LEFT—Sectioned '32 grille holds chromed wire mesh grille, sits above a chromed '48 Ford front end. Handmade brackets hold accessory headlights. LOWER LEFT—Rear Panel hides a '53 Ford rear end located by use of '49 Olds radius rods. '48 Olds shocks snub action of wide base Lincoln wheels and racing slicks. BELOW—'56 Chevrolet dash was narrowed 8 inches, special steering wheel from sheet stock. Both doors welded shut with white Naugahyde interior. RIGHT TOP—Handsome roadster was built by cutting the top of a '27 'T' coupe body, channeling the body over a '31 Model A frame, completely boxed for strength. RIGHT BOTTOM—'48 Ford steering lurks beneath stock displacement '56 Chevy mill. Iskenderian cam and lifters help pass fuel from six-carb Edelbrock manifold through polished ports to injector type pistons. Flywheel and clutch are Corvette, transmission '37 Packard.

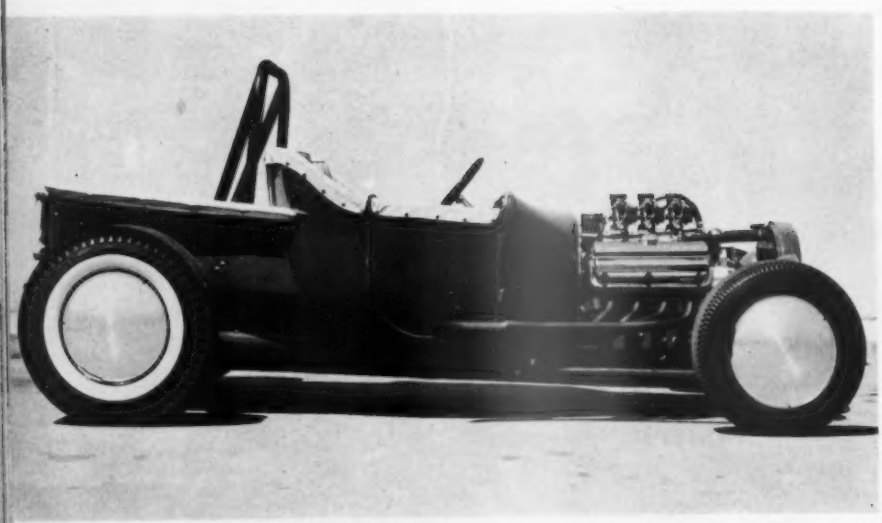
HOT ROD SHOWCASE



PHOTOS BY BOB HEGGE

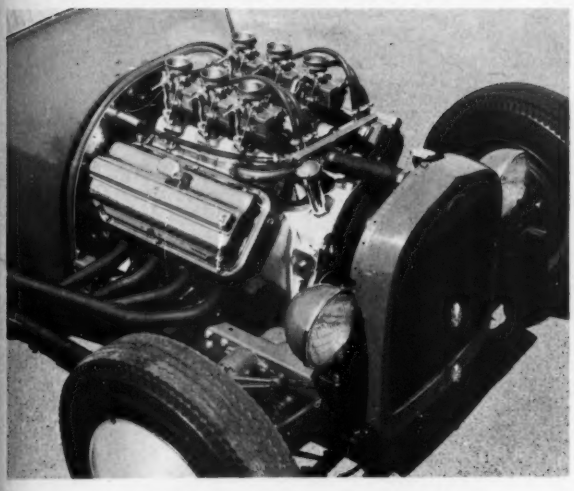


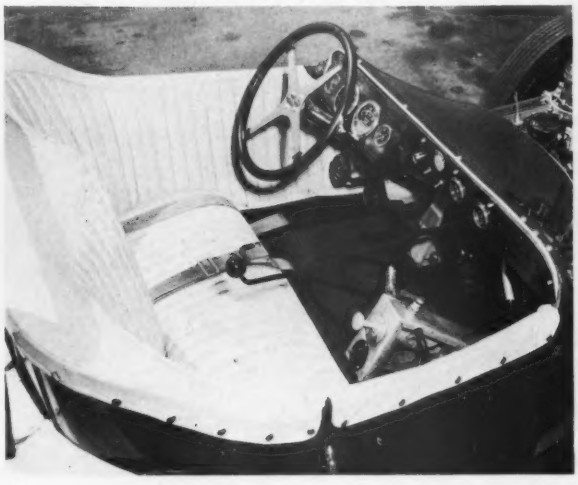
FAMILY ROADSTER

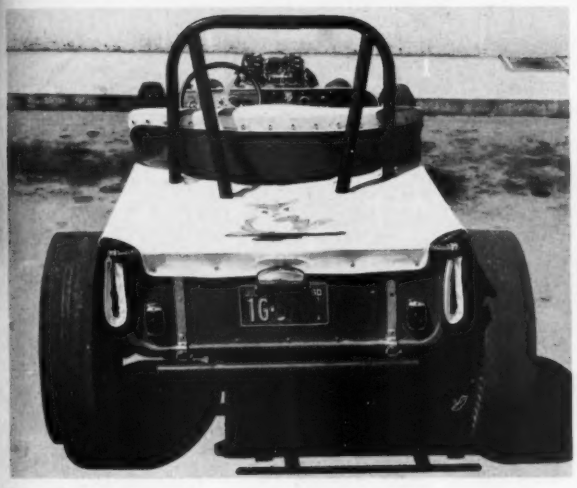


TOP—Bernice Wusz, right, sizes-up roadster that $700 baby-sitting funds helped build. Daughter Karen financed chrome plating for car while Tim and Father Frank Wusz built truck. J. Miller helped on mill.

LEFT—One of this rod's finer features is smoothly rolled side pan below frame to provide solid appearance, cover miscellany. Chassis, headers, roll bar painted dull black; body Aztec Red. Cost, $1500. Although no speed times were revealed by Wusz "company" it is assumed that their rapid red roadster pickup storms the quarter-mile road in extreme haste.







PHOTOS BY PETE SUKALAC

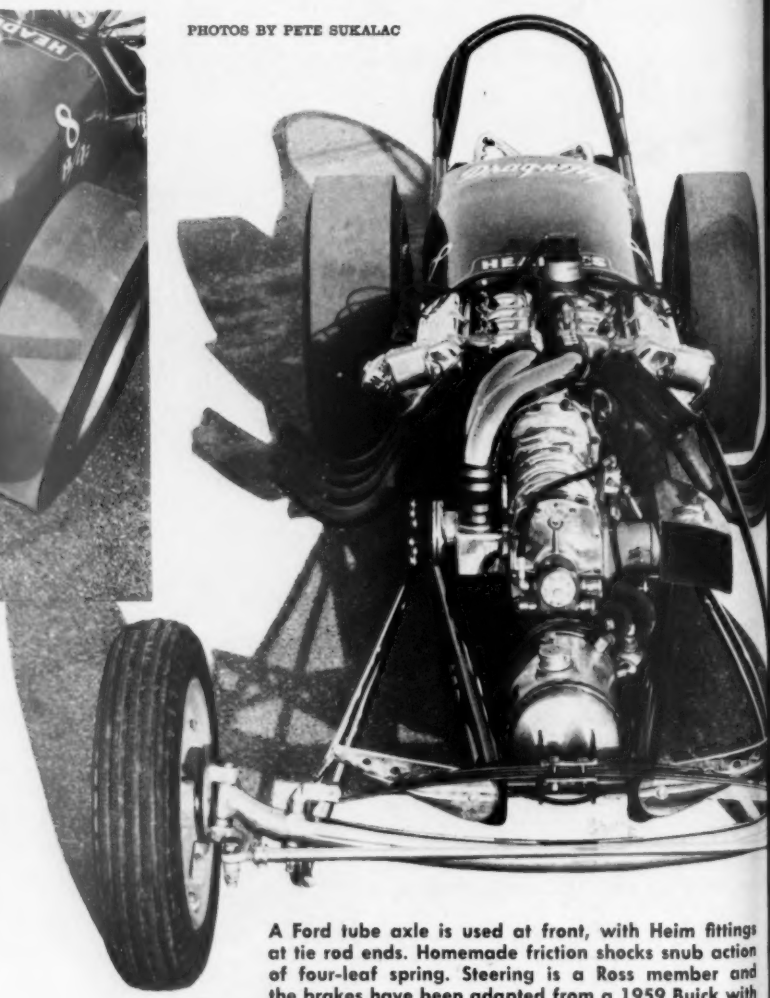
TOP LEFT—341-inch DeSoto has full Iskenderian cam kit, Jahns pistons and Sealed Power rings pumping 11½ to 1 compression. Thirsty V8 draws upon six Strombergs carried on Weiand manifold. Ignition is Spalding Flamethrower; flywheel is 16-lb. Drag Action. TOP RIGHT—Cockpit is snug but comfortable. Contoured seat is cleverly shaped to snap over edge of body for more rounded effect. Fuel pressure pump, gauge and shutoff valve in near-side unit. Steering is '34 Ford. CENTER LEFT—Bracketed by 7.60x15 Bruck slicks are Chev lights and simple nerf bar. CENTER RIGHT—Stubby Dodge looks pretty wild. RIGHT—In reality this is '23 Dodge touring cut down and the frame is Model A.

ALL 'ROUND RAIL

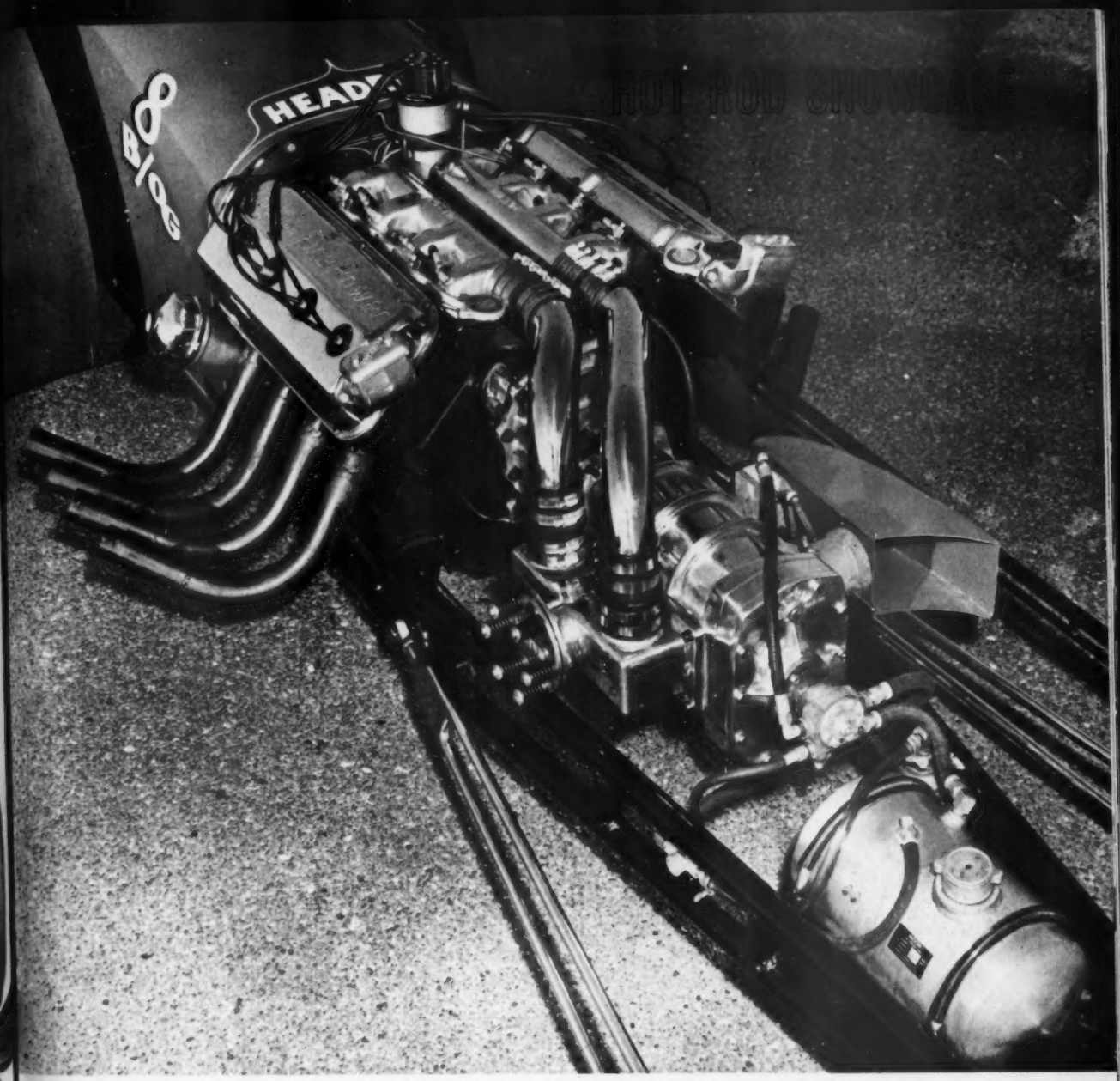
Dale Greer built the tube truss frame to 101-inch wheelbase, selected Halibrand wheels to mount Firestone tires in front, M and H slicks at the rear. The long drag link has a secondary brace welded below it to eliminate whip and has been fitted with Heim ends to insure absolute control of the front end. The body section is formed of one sheet of aluminum held in place by machine bolts with the area about the adjustable pitman arm hammered to clear the arm's fore and aft action.

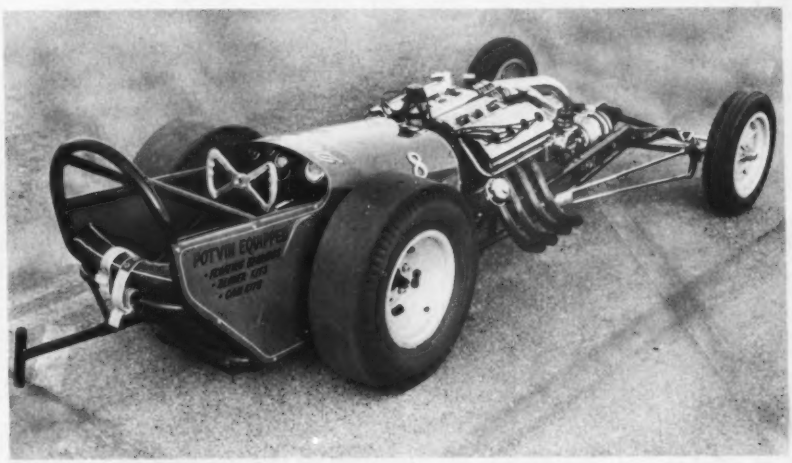
PHOTOS BY PETE SUKALAC

Pilot's office is upholstered in black and white Naugahyde, encloses Halibrand quick-change unit using 3.78 gears. Schiefer flywheel, pressure plate and Velvetouch disc pass torque through '37 Cad box to 40-inch wide Ford rear end. Dale Greer built this dragster in seven months and set a record of 132 mph in 10.74 second in Bayview, Washington. This is the latest addition to the stable.



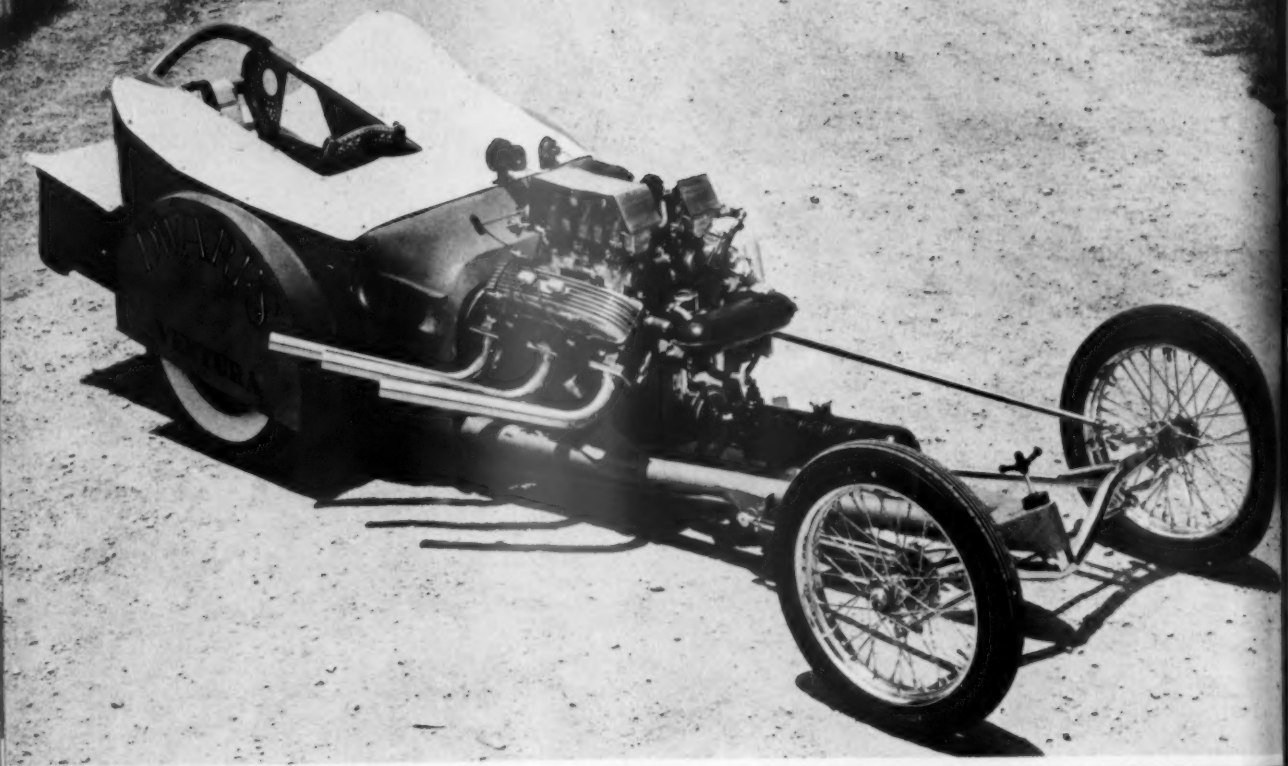
A Ford tube axle is used at front, with Heim fittings at tie rod ends. Homemade friction shocks snub action of four-leaf spring. Steering is a Ross member and the brakes have been adapted from a 1959 Buick with aluminum drums. Best performance of this car to date with limited outings, 142 mph in 10.56 seconds.

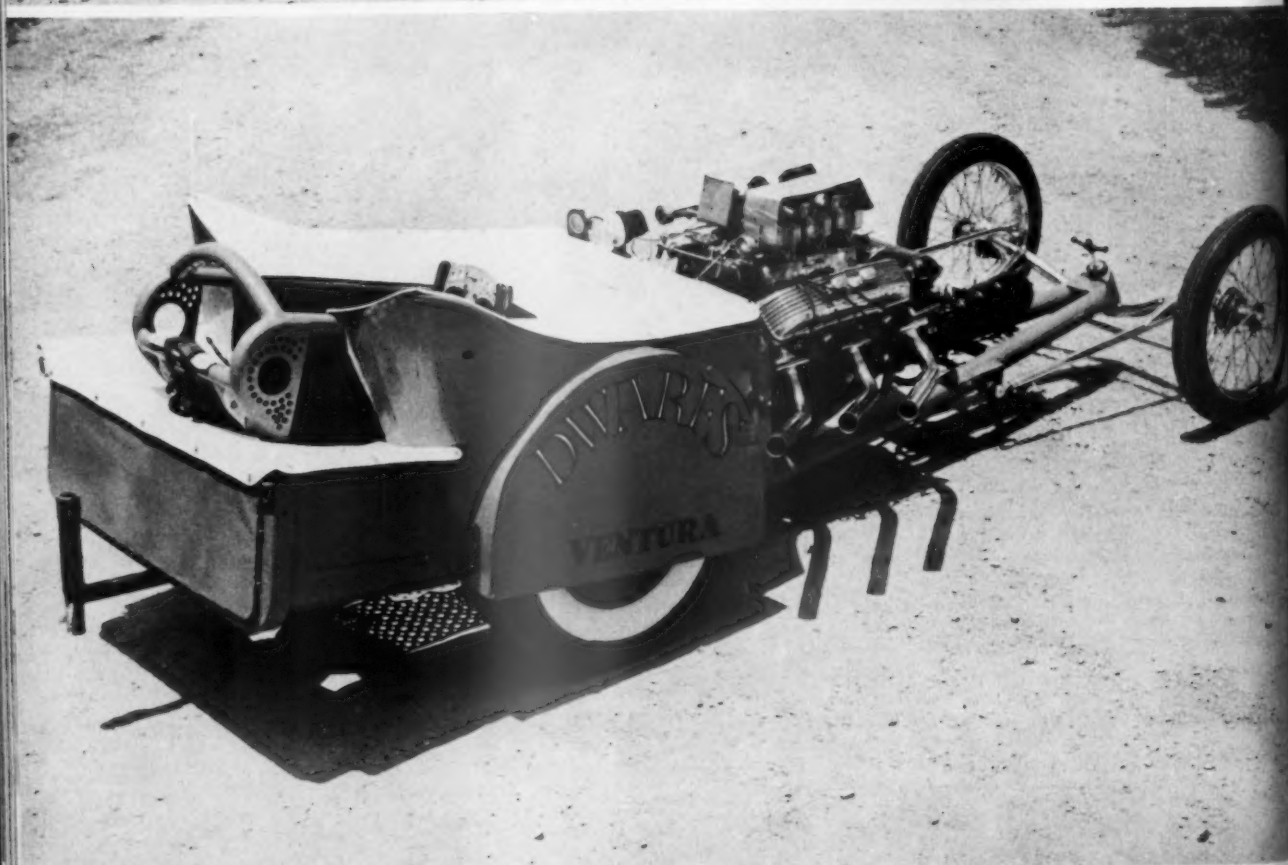




Detailed inspection shows the painstaking workmanship on Greer dragster. The 331 cubic inch '55 Chrysler engine carries a Potvin cam kit, Forgedtrue pistons, Potvin blower kit, Scintilla Vertex magneto and Grant piston rings. Dragster set Bozeman (Mont.) strip record of 128 mph at 11.75.

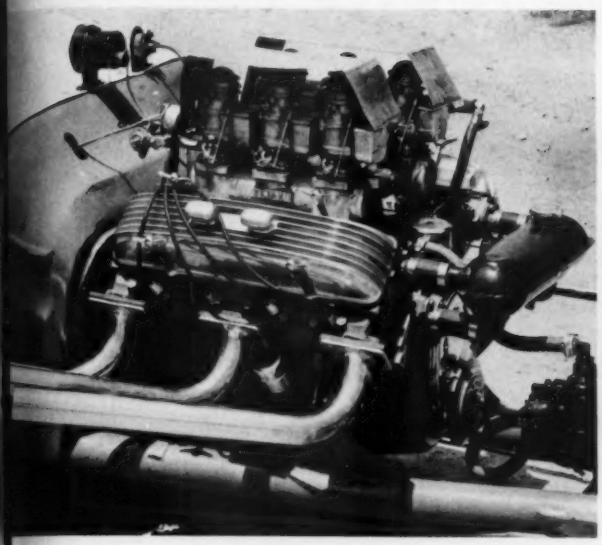
ROADSTER "RAIL"

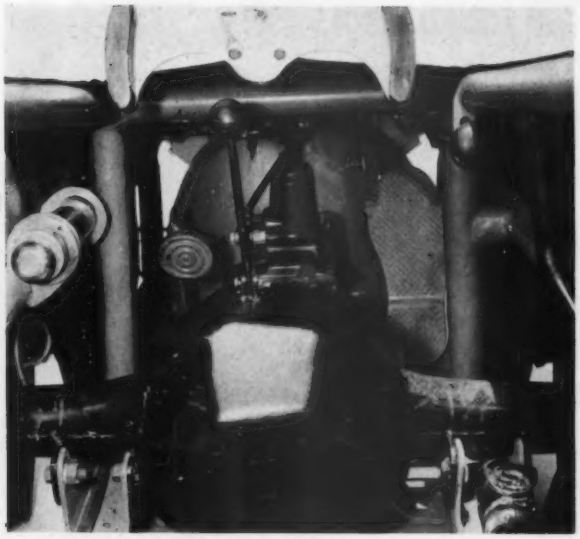




HOT ROD SHOWCASE

PHOTOS BY CURT HAMILTON

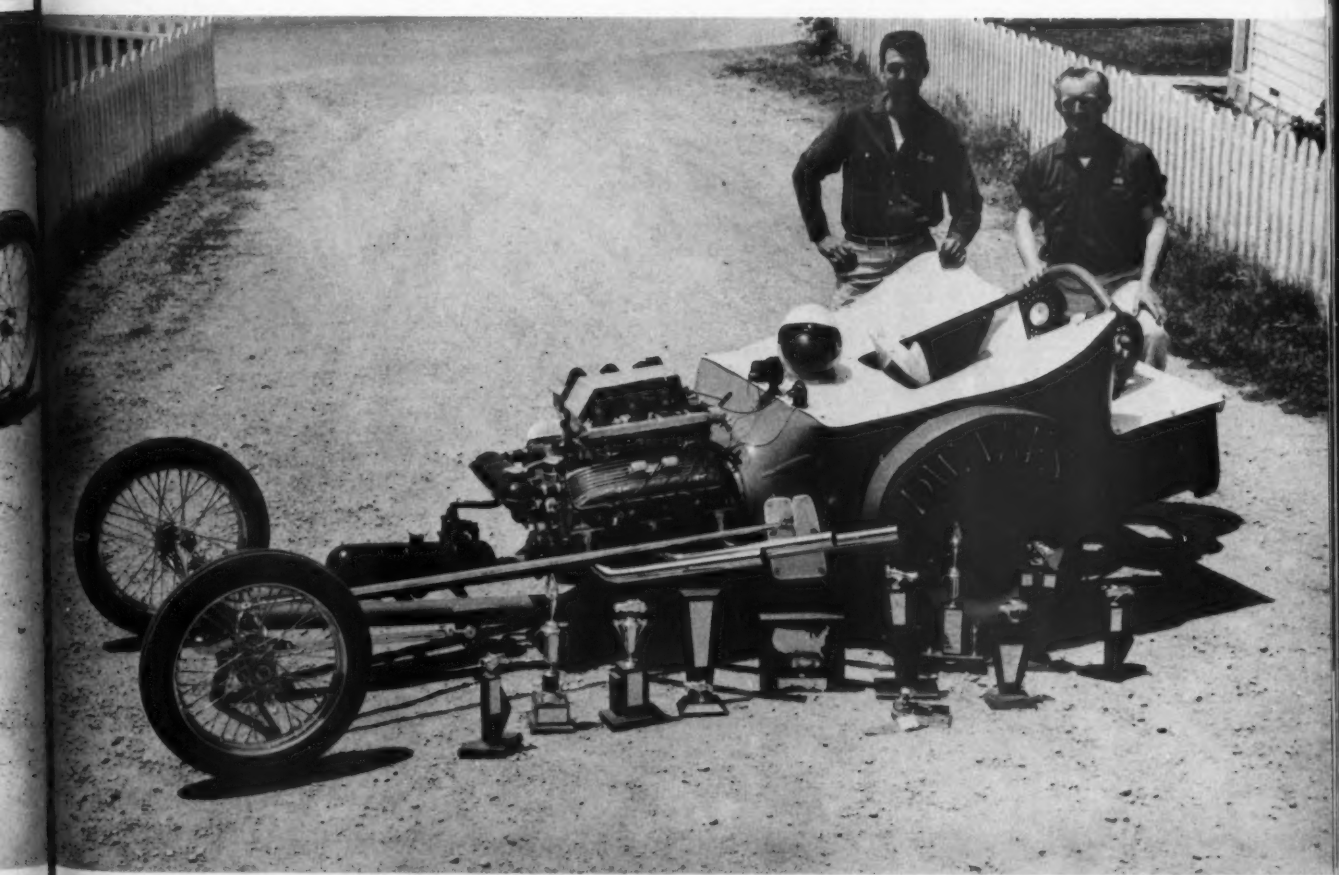




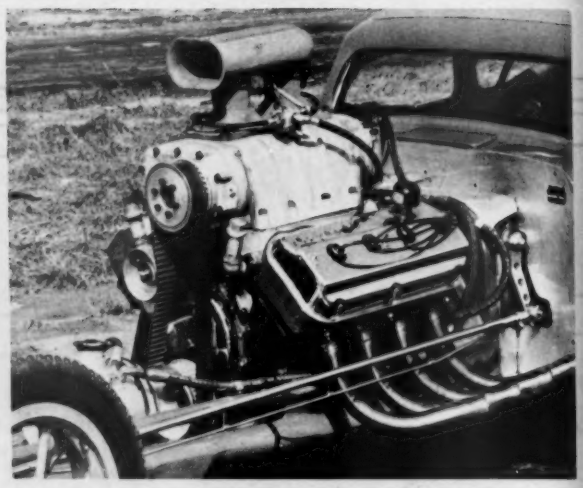
ABOVE—Carburetor City, with a population of six, rests on Weiand dual log manifold feeding 365 cubic inch '58 Cad V8 of stock bore-stroke. A Chet Herbert roller cam setup is featured inside with stock Cad pistons, Perfect Circle rings.

BELOW — Billy Bridges and Garland Moody with car and victors' hardware. Their quarter-mile rig has 96-inch wheelbase, 50-inch front tread, 42-inch rear. A Schiefer aluminum flywheel-pressure plate is run with 11-inch Velvetouch disc.

ABOVE—Keeping the 3.90:1 Olds differential gears out of the driver's lap is a nearly full enclosure. Remote gear shift operates '39 Ford box with second and high gears only. Gearbox is adapted to Olds rear end with machined U-joint.

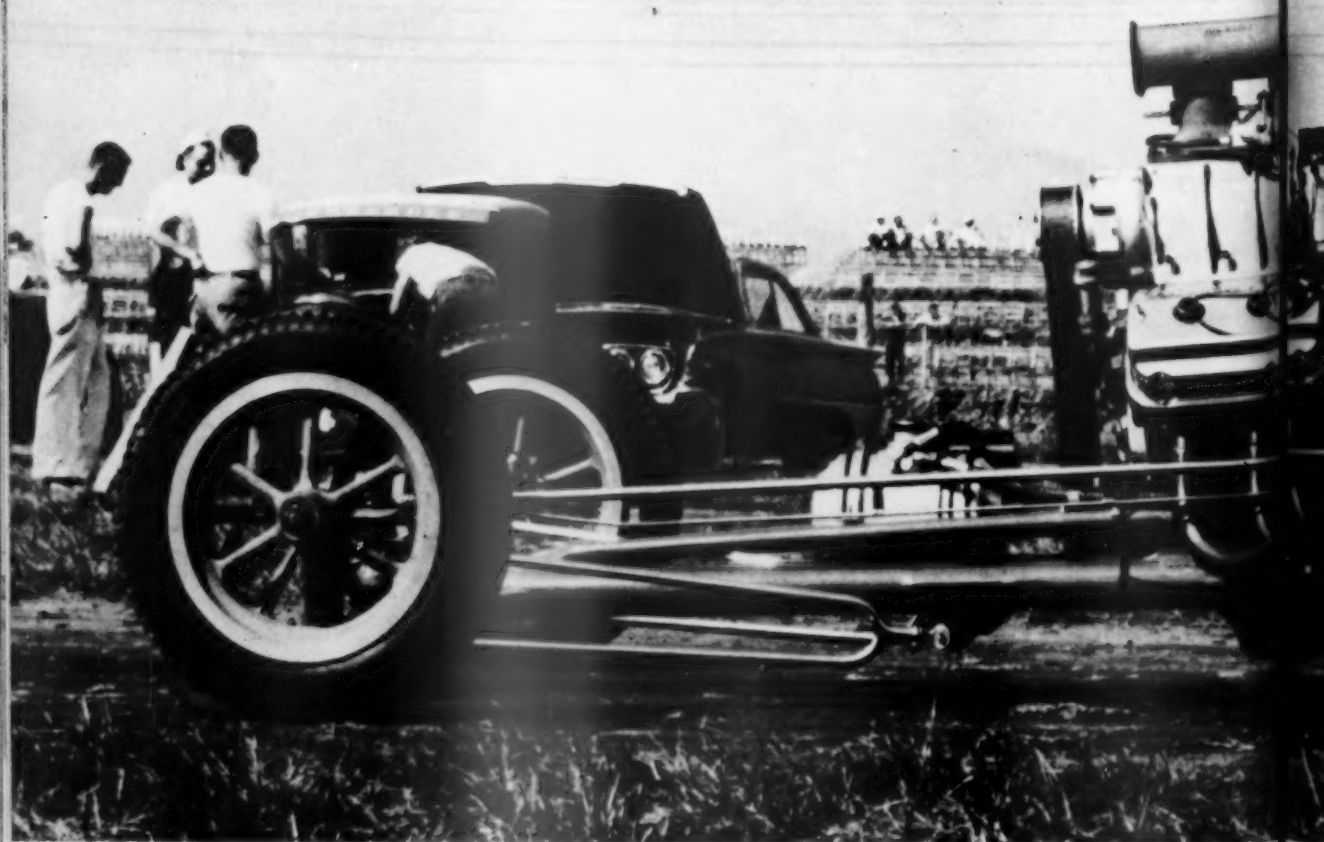


"AGGRAVATION"

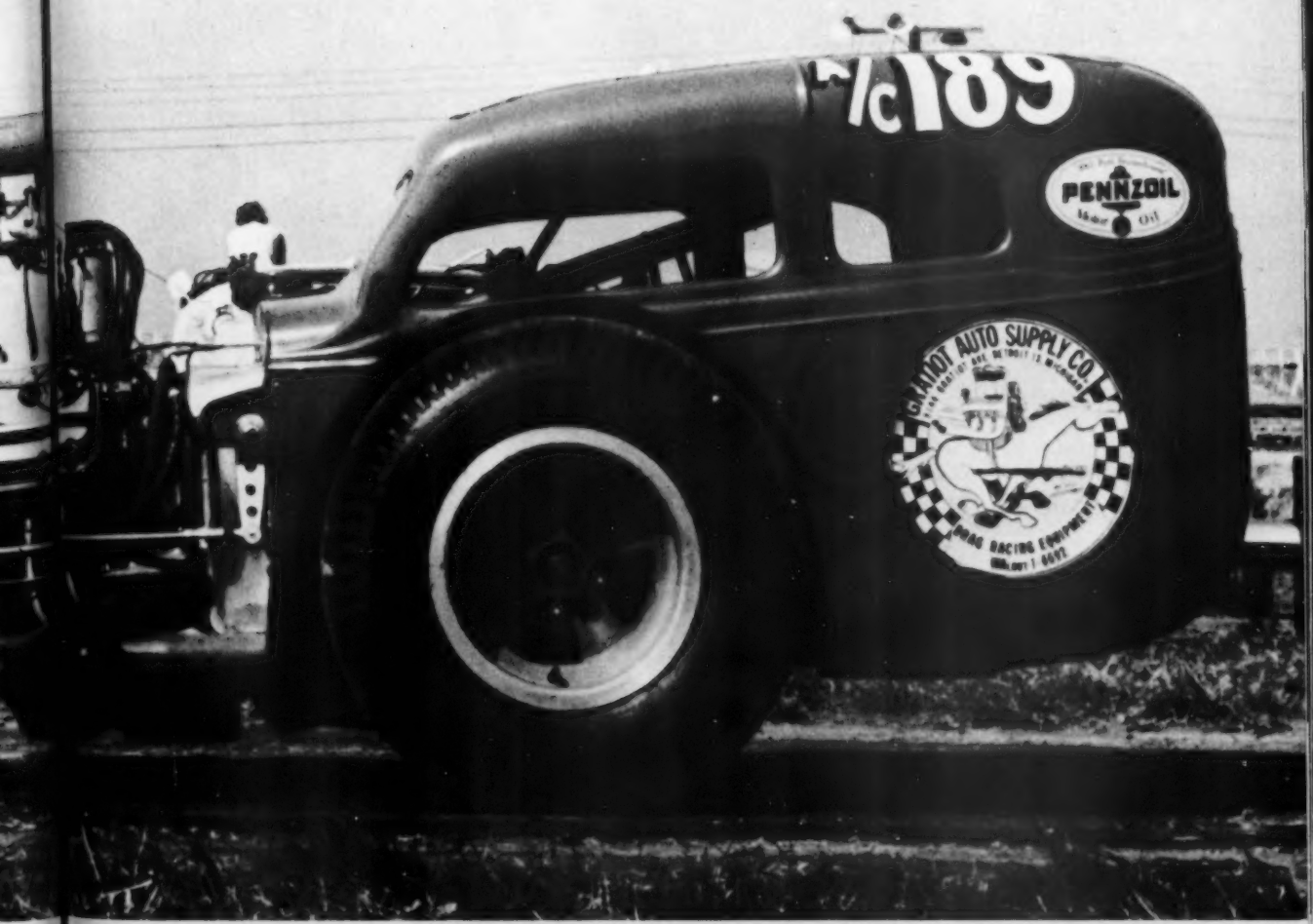


Al Bergler built car with help from Michigan Hot Rod Association buddies. Front axle is positioned by a short spring, friction shocks, and reversed spindles to place tie rod ahead of axle for freedom of movement. Big 400-inch Chrysler gets full pumping action from 6-71 GMC blower with Hilborn injection system atop Weiand intake manifold. Valve action is handled by a Harman & Collins cam and lifters, home-made pushrods and Gotha rocker arms. Thompson pistons, Vertex magneto are also used.

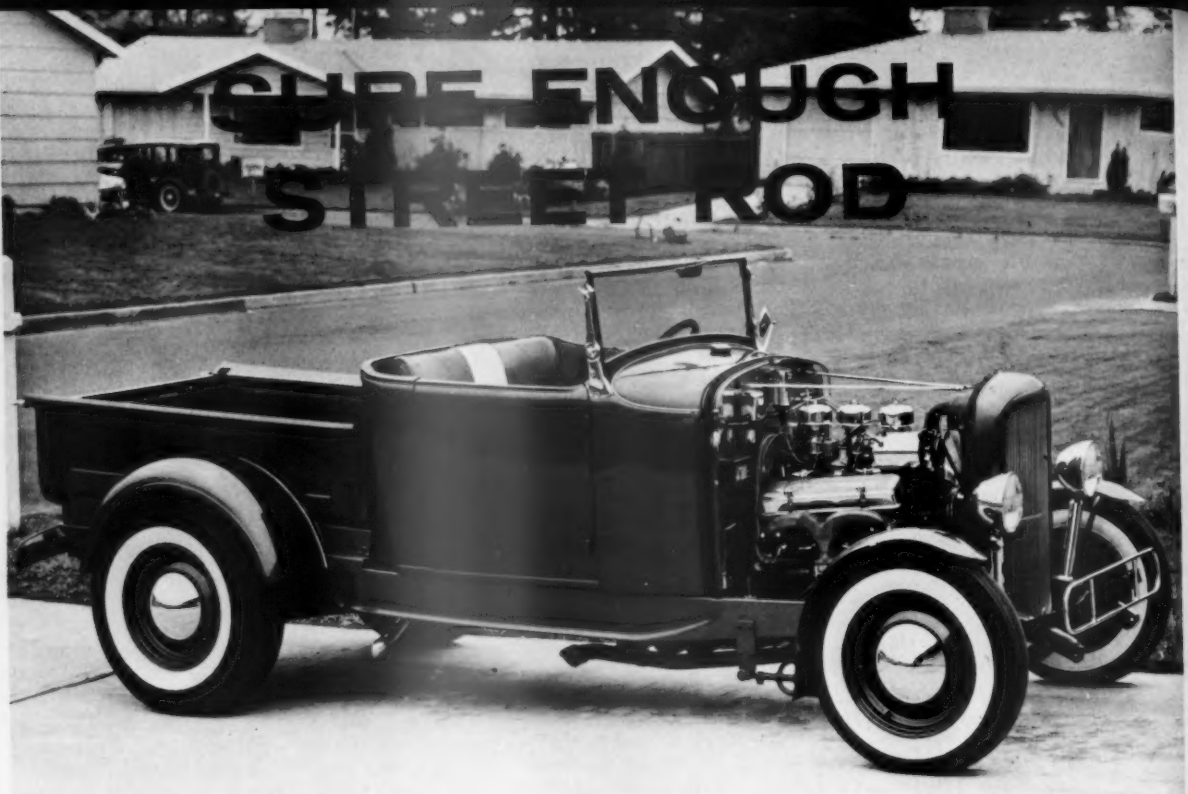
PHOTOS BY ERIC RICKMAN



1960 National Drags found Bergler's bomb on the line but the car was a bit too new to steal any titles from the veterans in its class where speeds up to 160 were posted. Aggravation's best attempt at that time was 135 in the quarter but thoroughness of construction indicates bigger things are to come in the ensuing warm-ups at the car's home track, the Detroit, Michigan Dragway.



SURE ENOUGH STREET ROD



HOT ROD SHOWCASE

PHOTOS BY PETE SUKALAC

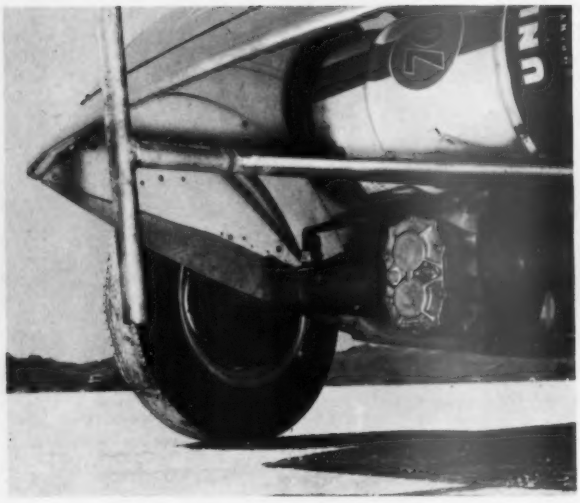
LEFT—'30 Ford pickup body and box are completely stock but now rest on a pair of '32 frame rails. Front fenders are cycle while the rears are bobbed Ford. Wheels have '48 Ford centers, '52 Lincoln wide-based rims, 15-inch tires.

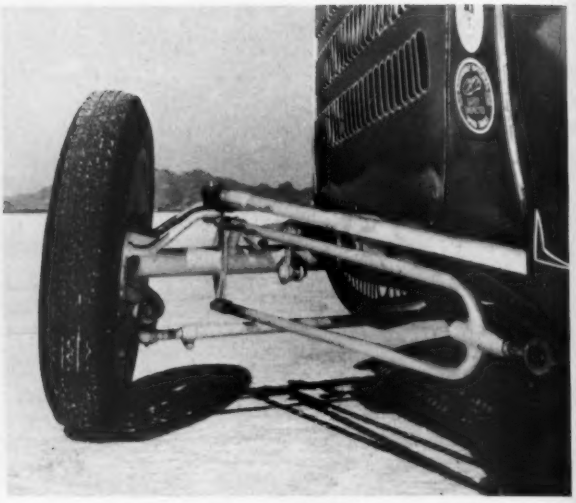
BELOW—265-inch Chevy engine has been fitted with an Isky cam, '58 heads, and Offenhauser intake manifold; it's stock otherwise. Coupled to a Powerglide transmission, power output is more than enough to keep the 'A' ahead of traffic.

ABOVE—A transplanted southerner, Hamer Phillips, designed his rod for practical everyday use and shares the driving duties with his wife. Note that fuel tank remains in stock location under cowl with an electric fuel pump booster.

ABOVE—The cockpit has two-tone blue Naugahyde interior, blue carpet. Model A instrument cluster is kept but fitted with S-W gauges. Steering wheel is a Bell item while floor stick selects gears for a '55 Chevy Powerglide transmission.

DOUBLE DUTY CHARGER





TOP LEFT—Ford rear axle with Halibrand quick-change center section bolts solidly to the '29 Ford frame. Large tank is for water, rear mounted for traction.

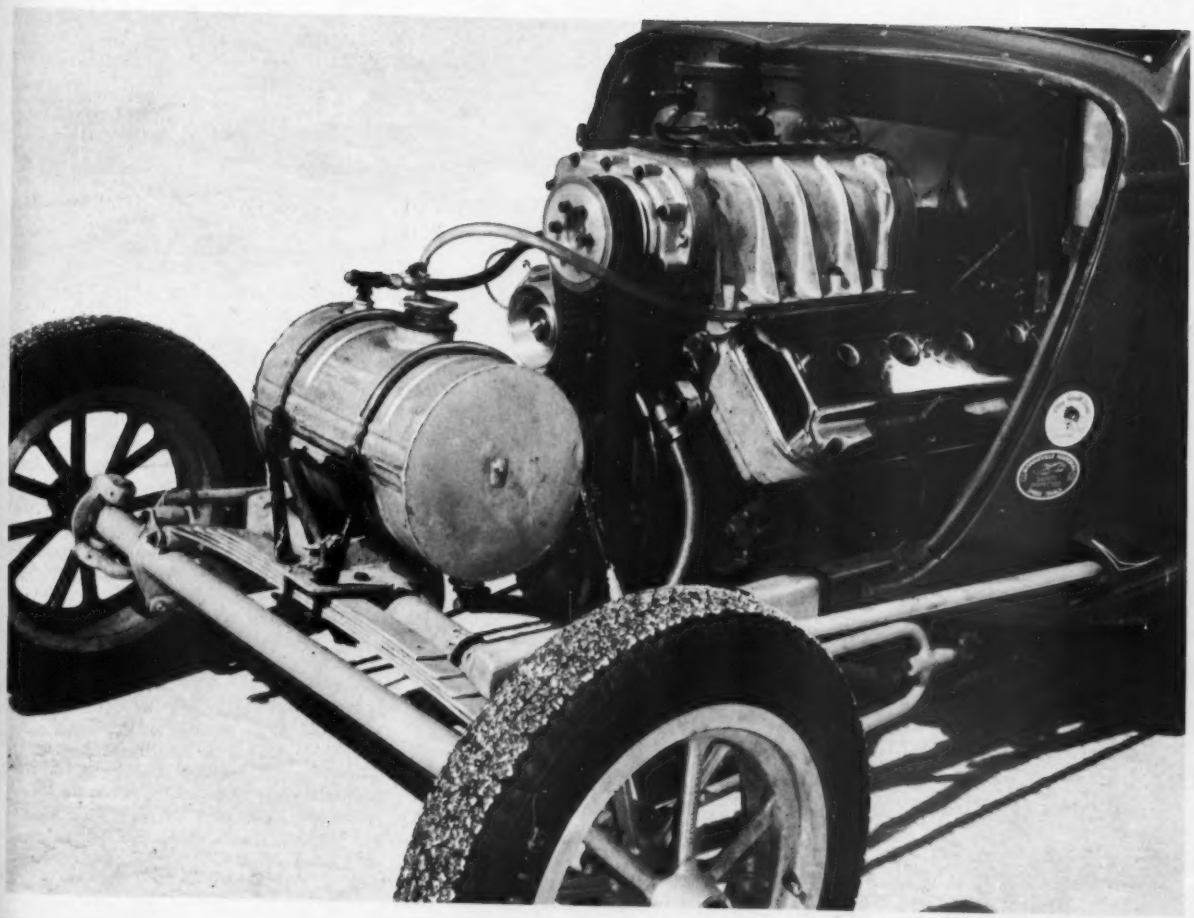
ABOVE—Brakes are not used up front for this straightaway coupe. Clevis ends on front of the radius rods allow wide caster adjustment while threaded Heim bearing end at rear permits exact wheelbase setting. Parts are metal sprayed.

LEFT—Owner Rich Richmond atop storm door as he prepares to push off for a trial run at the '60 Bonneville Nationals. The front axle is homemade with Ford spindles and friction type shocks.

RIGHT—331-inch Chrysler has a Clay Smith cam, Hilborn injection, Forgedtrue pistons, Vertex mag and a 6-71 blower. This engine powered gas roadster to 188 mph record in '59 but blew up this year.

PHOTOS BY DON FRANCISCO

HOT ROD SHOWCASE

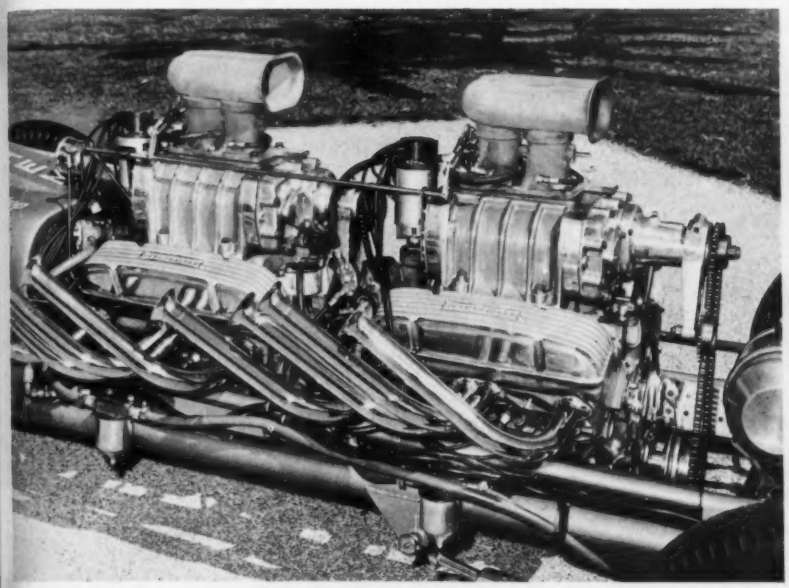


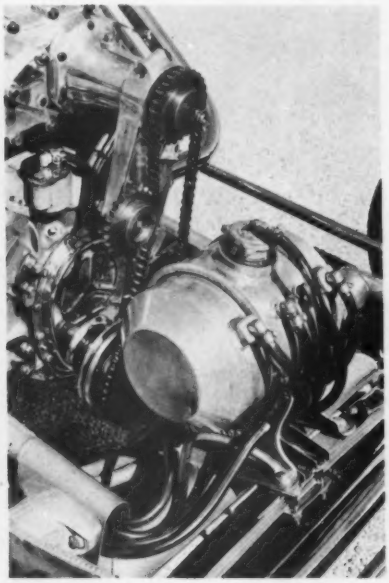
"MAYHEM"

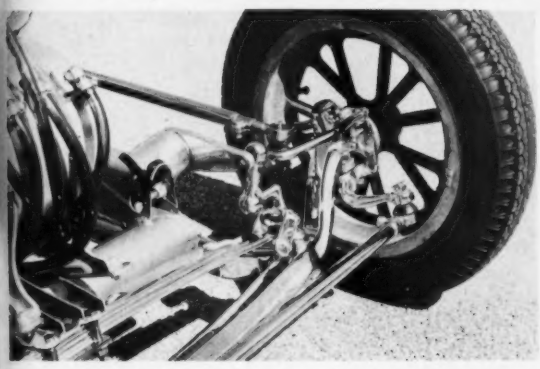
Dubbed "Mayhem" for its calculated attitude toward competition, this double Chevy dragster was made for the purpose of capturing NHRA gas record, was built in just one month, cost $7000.

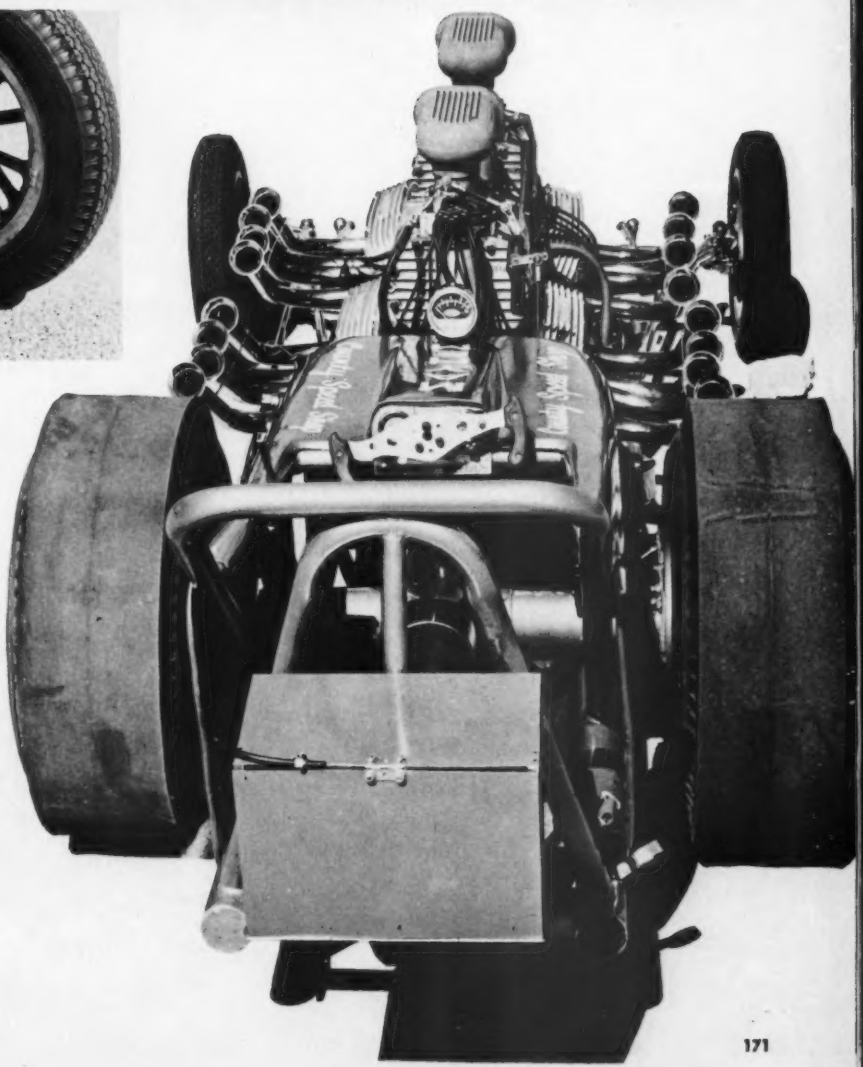
LEFT—Ed Fitzwilliam's dragster is as attractive as it is fast, taking safe construction awards in stride. He built it around a '60 model Chassis Research frame, using 104-inch wheelbase, 56-inch front tread and 48 rear. Rear axles are '48 Ford parts using Halibrand center section with 4.11 ring, pinion gears.

PHOTOS BY ROBERT HEGGE



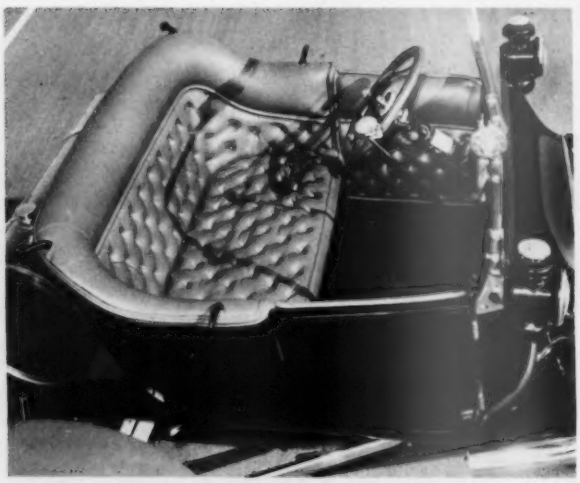


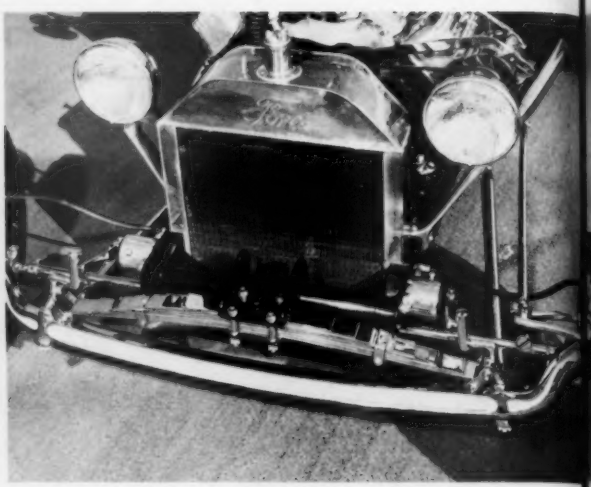


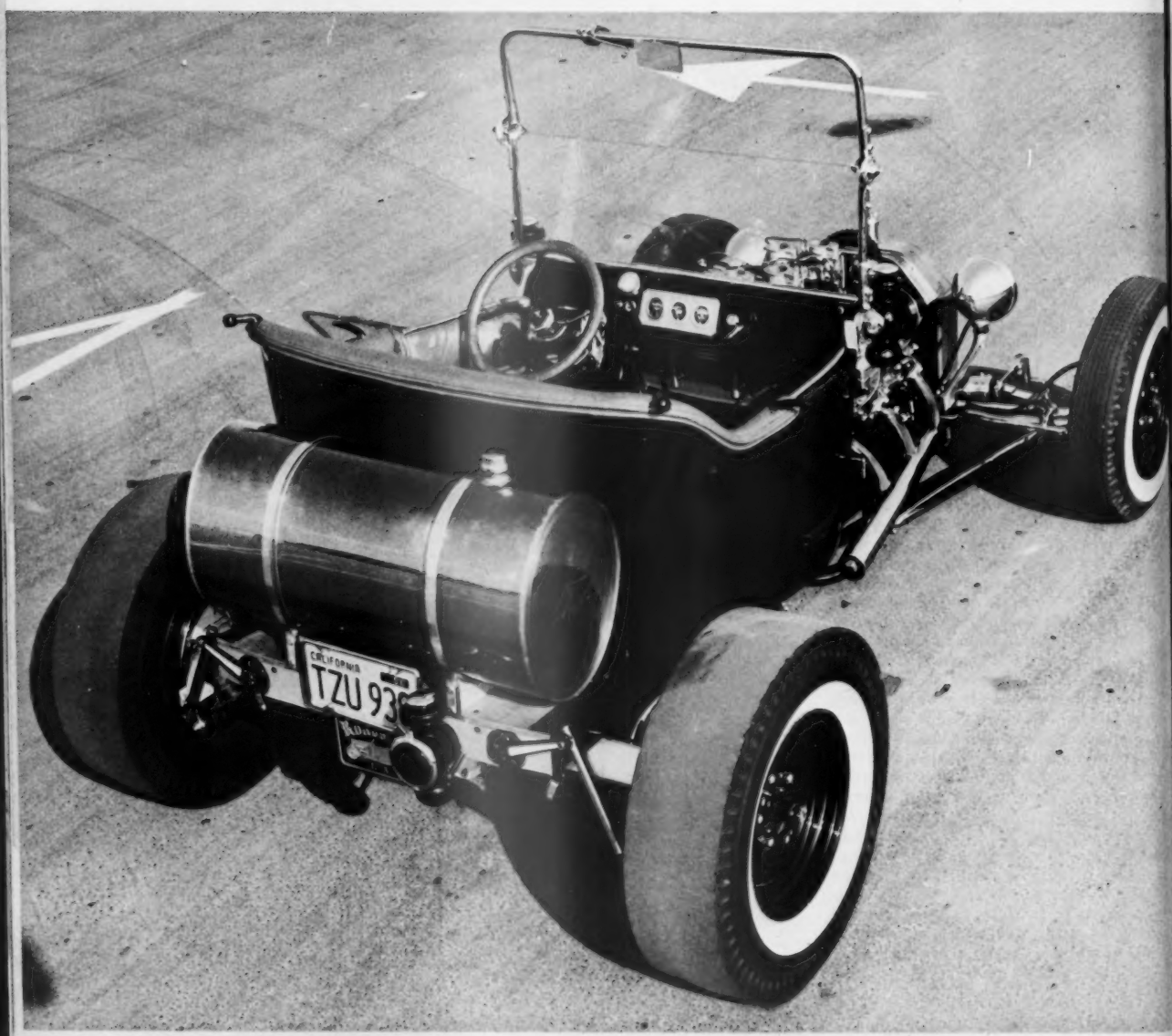


680 collective cubic inches of screamin' Chevy spin big GMC superchargers topped with Hilborn injectors. Blowers mount on Reath manifold. Chain guards for blower drive were removed from these photos. V8's are punched out to 4 inches, stroked to 3⅜ inches, run Isky cam and lifters, Chev pushrods and rockers, and Forgedtrue blower pistons with Sealed Power rings. Ported, polished heads cap 8½:1 c.r. Taking the thrust is a Schiefer 11-inch pressure plate, dual disc and a Crower button flywheel on rear engine. Ahead of the blower drive sits a compact fuel tank with large diameter feed lines emerging from bottom of tank and running to injector pumps for respective engines (incorporated with mag drive at rear of engines). Smaller lines into top of tank are fuel by-pass return lines, plus pressure relief line. Note routing. Up front is '48 Ford beam axle dropped 4 inches. Note that wheel spindles were reversed left for right to place tie rod ahead of axle, eliminating interference with frame during movement of axle.

STREET 'T'







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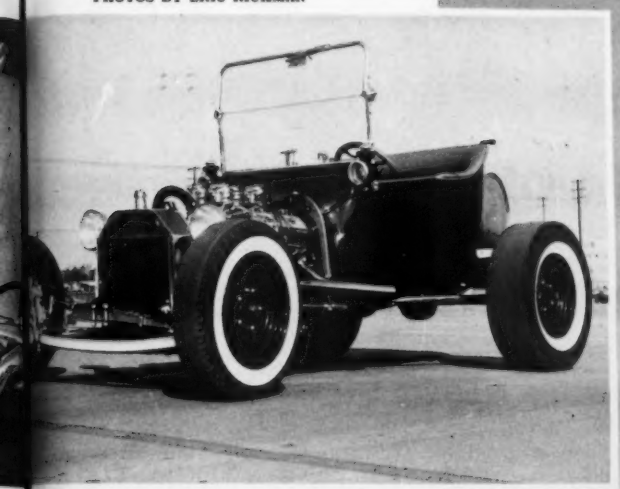
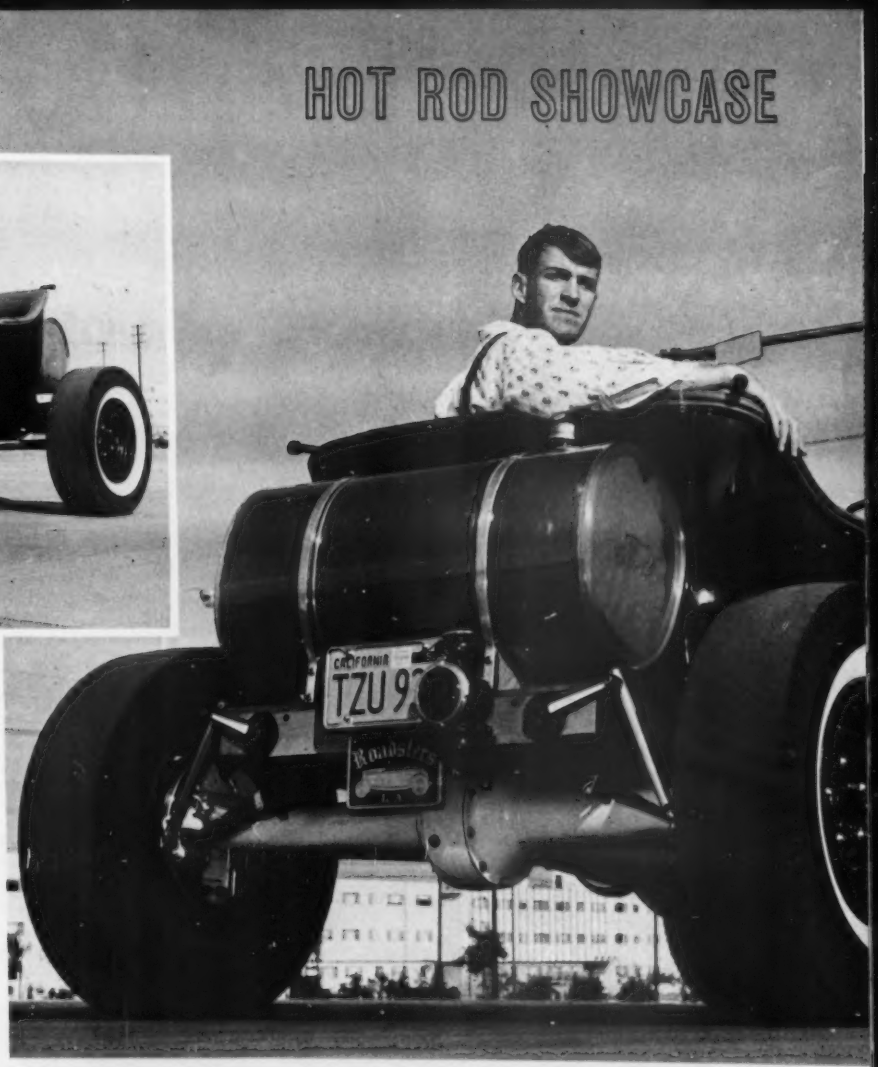
HOT ROD SHOWCASE

PHOTOS BY ERIC RICKMAN



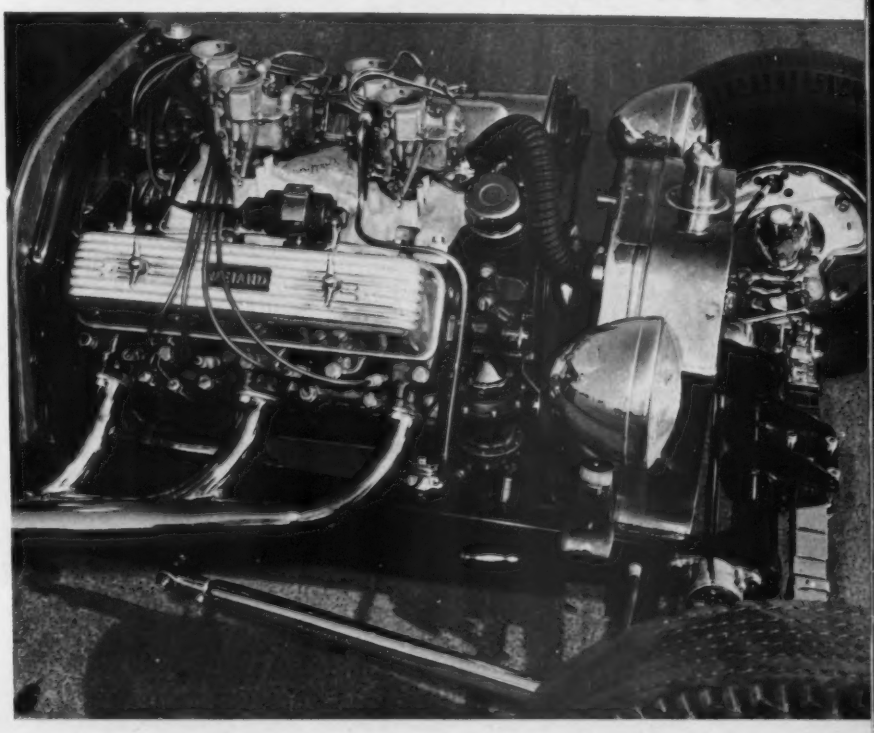


FAR LEFT—Beautifully tufted Naugahyde upholstery is the handiwork of Ken Mulhern at Dorn's Norwalk Trim Shop. This type of upholstery design, along with original 'T' steering wheel, finds great favor with western rod builders. Steering gear in Martin's car is '48 Ross.

ABOVE LEFT—Closer inspection of front end detail reveals Dietz headlights, truck brake hoses, split radius rods and dropped Ford beam axle with leaf spring.

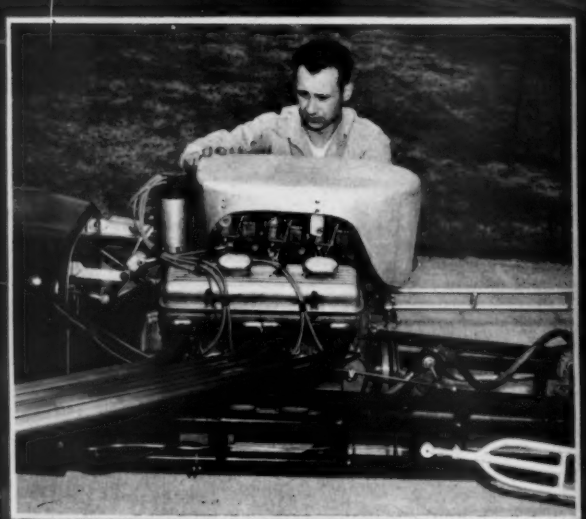
ABOVE—Marty Holmann's latest street roadster is this 1915 'T'. Body work on the jet black jewel was handled by Mike Scott while Don Prudhomne applied the enamel paint. Chrome by Valley Plating.

LEFT—Stock height windshield is just right to protect occupants from wind blast. Wood paneling provides base for brass gas tank; taillight and parking lamps are classified as antique items. Martin gives much of the credit for building his neat roadster to pal Darryl Sutton.

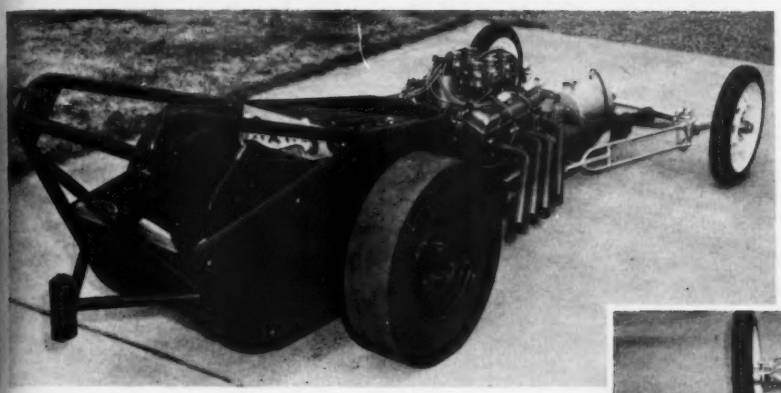


RIGHT—Holmann's rod is powered by a '49 Olds of 324 cubic inches running a '55 Olds cam, collapsed lifters, Thomas rockers, American Hammered rings, '56 Olds pistons, milled heads and an Edelbrock manifold. Ignition is by Mallory.

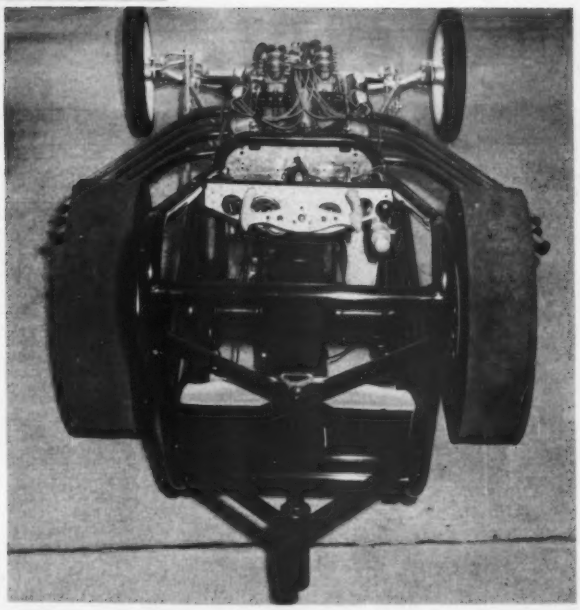
SPEEDY AND SAFE



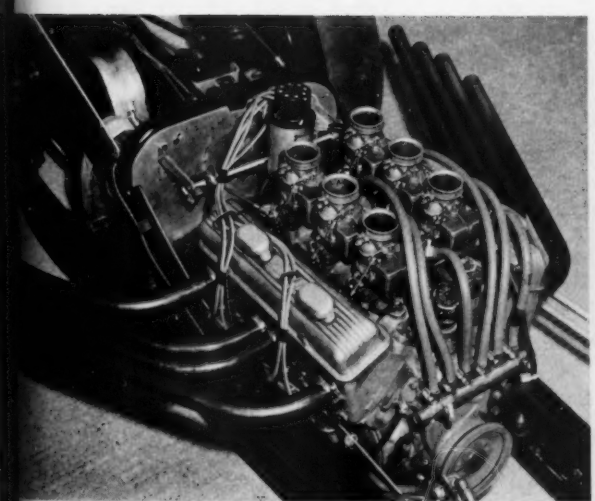
PHOTOS BY PETE SUKALAC



Curve in roll bars is braced with drilled gussets. 8.00x15 M&H drag slicks mounted on widened 15-inch Chevrolet wheels.

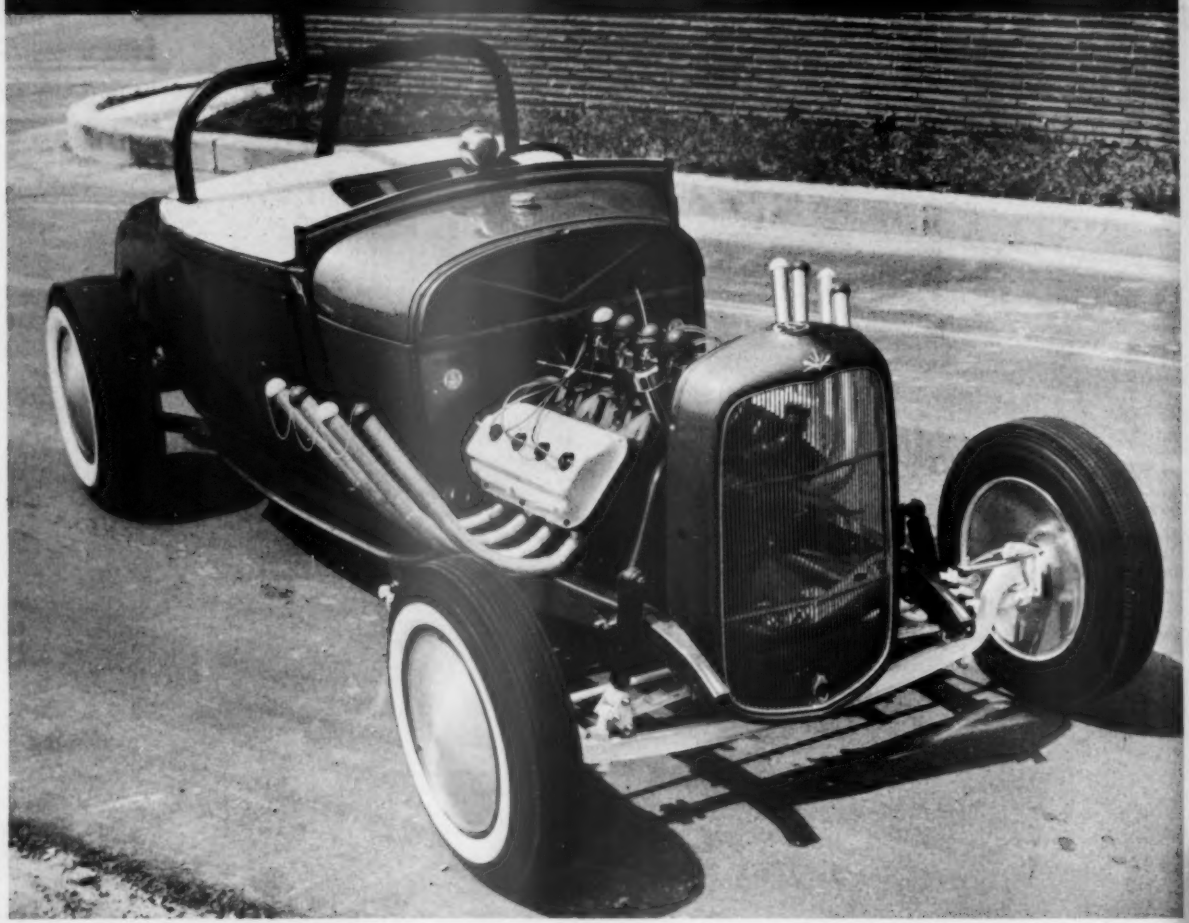


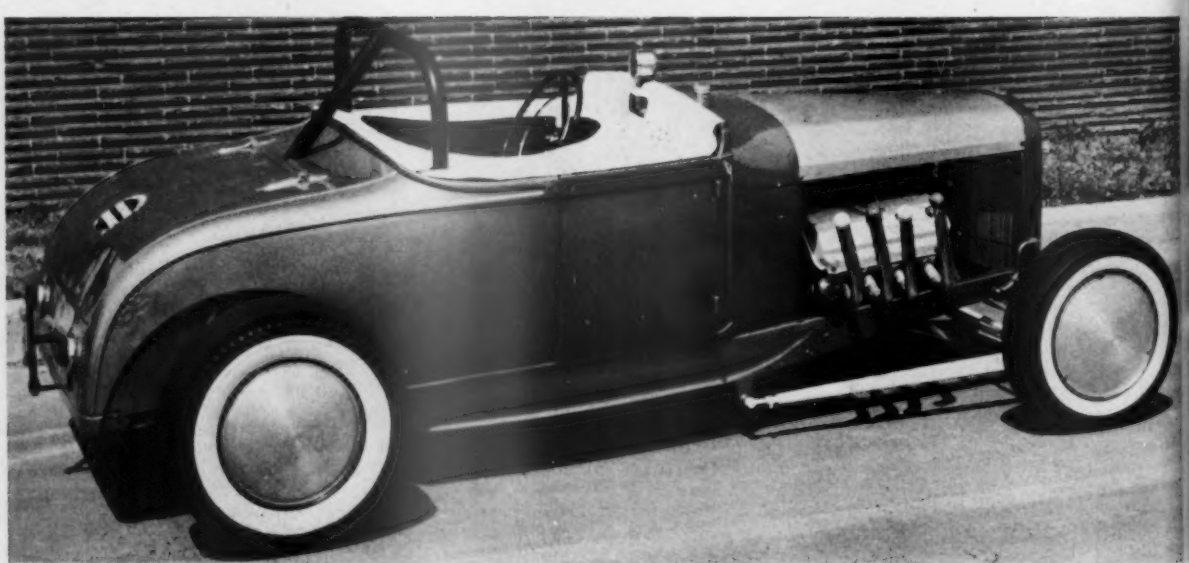
NHRA Technical Advisor Chuck Blanchard is a firm believer in absolute safety, but he also believes in speed. His 1410 pound rail turns 142.80 mph in 9.6 seconds in B class, drops weight to 1385 pounds with mag wheels, adds blower for A class speed of 155 mph in 9.5 seconds. Full carb shroud keeps oil and gas off goggles. Cage roll bars give Chuck maximum protection.



Above left: Blanchard first started on Chevys with an inline six 10 years ago. Present engine is bored to four inches with a half-inch stroke. 352-inch mill has Iskenderian 5 cycle cam & kit, Jahns solid skirt pistons, modified Corvette heads. Above right: Steering is Franklin. Chuck made his own steel flywheel, which holds an eleven-inch Hayes clutch assembly driving a cyclone quick-change rear end through a 1941 Cadillac side-shift gearbox. Early Ford rear axle housing is shortened to 40 inch width.

'29 SWIFTY

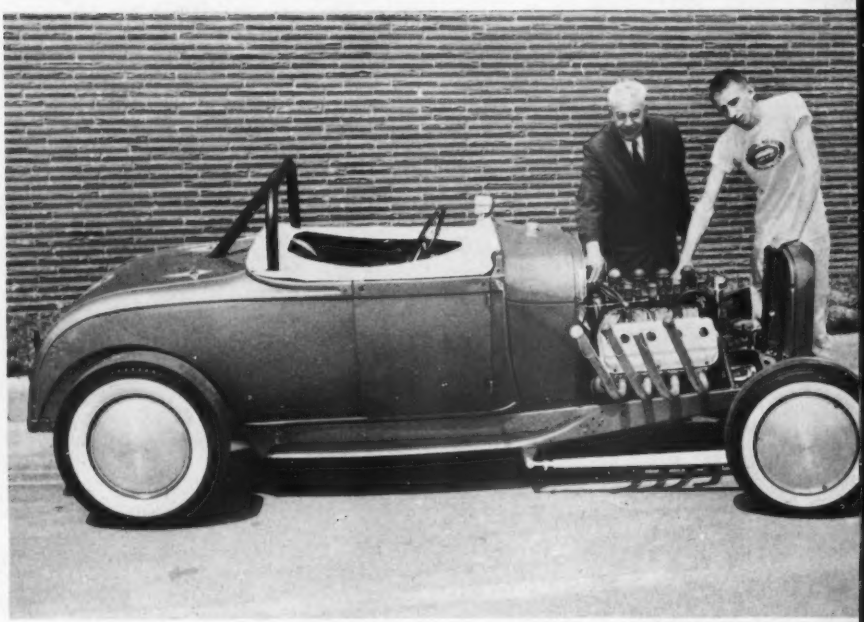




LEFT—A '32 Ford frame, Z'd 6 inches, carries Model A body and a muscular '51 Chrysler powerplant. Ford beam axle was dropped 3 inches, fitted with tube shock absorbers, side mount radius rods.

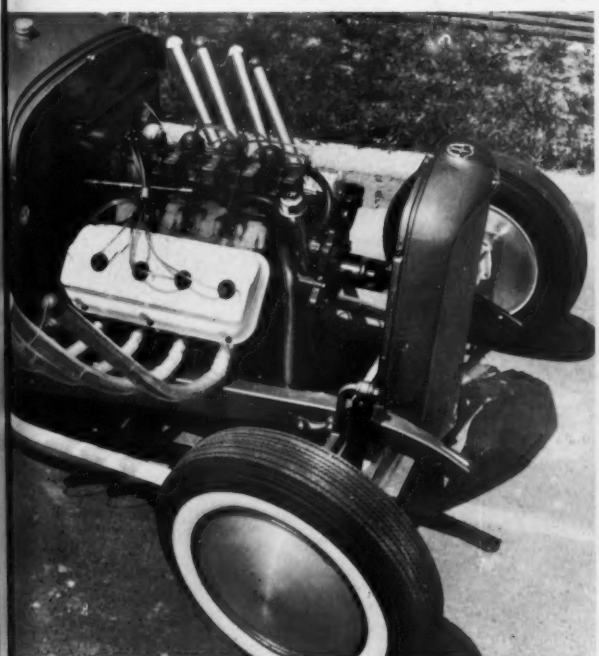
RIGHT—Hot rod and motorcycle enthusiast Lloyd Llewellyn didn't give up his pastime with the demise of the 'T' and the JD, he merely turned the reins over to the younger set. Lloyd, shown left, owns roadster; Henry Southworth and John Gotschalk drive and keep it tuned.

LOWER LEFT—The roll bar, push bar, open headers and 7:10 Bruce slicks give a clue to builders' draggin' intentions. Moon discs and Pontiac taillights add a final touch to Roadster class drag rod.

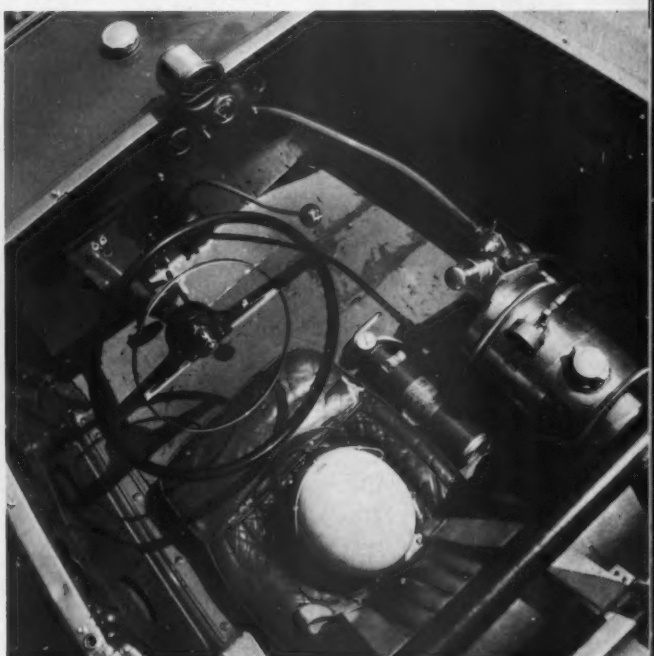


PHOTOS BY ED EATON

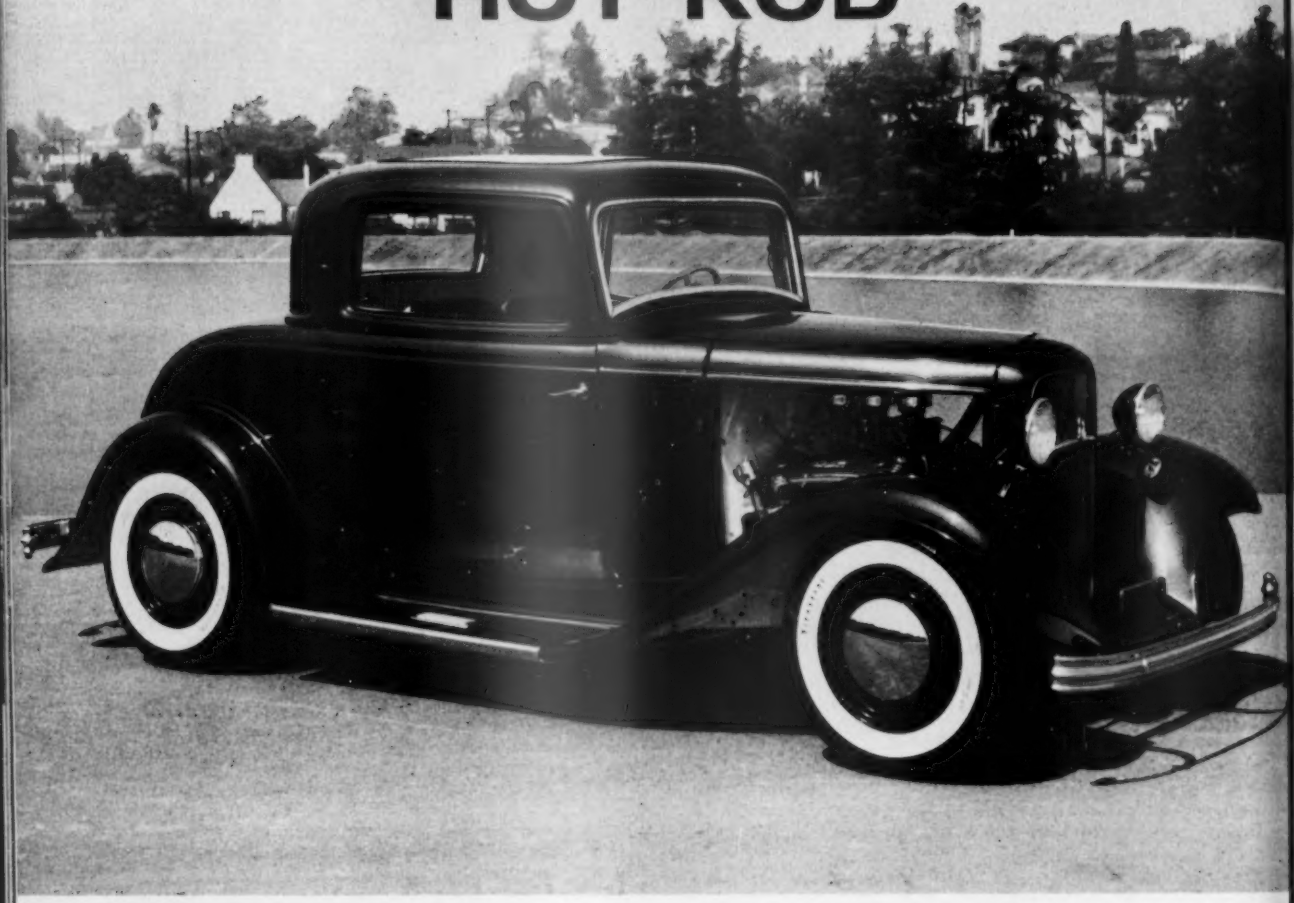
Chrysler is bored 1/8th with stock stroke, runs Isky 5-cycle cam and kit, Jahns 10:1 pistons. Ported and polished heads, lightened and polished valves are also featured inside. Bolt-on equipment includes Weiand 4-carb manifold and 97's with richer jets, and Vertex magneto. Moon fuel system is used.

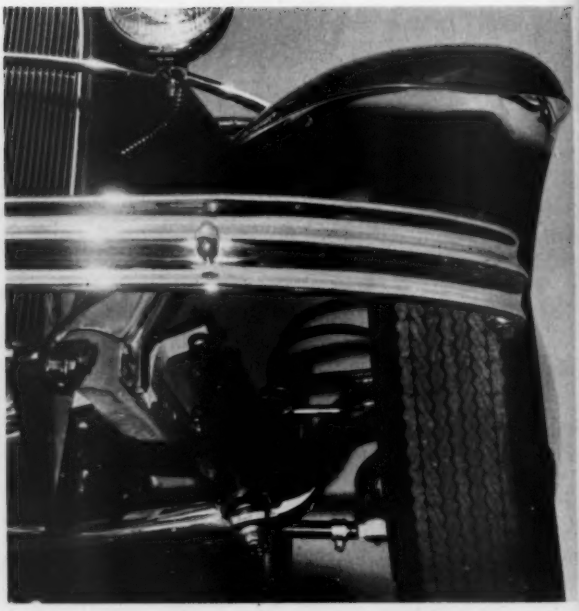


Oil pressure, tach and temperature gauges are easy to read, while fuel pressure pump and fire extinguisher are right at hand. Plush seat is from a '57 Jaguar. Below floor is a '39 Ford box running second and high gears in conjunction with 4.44 Ford rear end. Dodge truck clutch and flywheel are used.



"ASSEMBLY LINE" HOT ROD





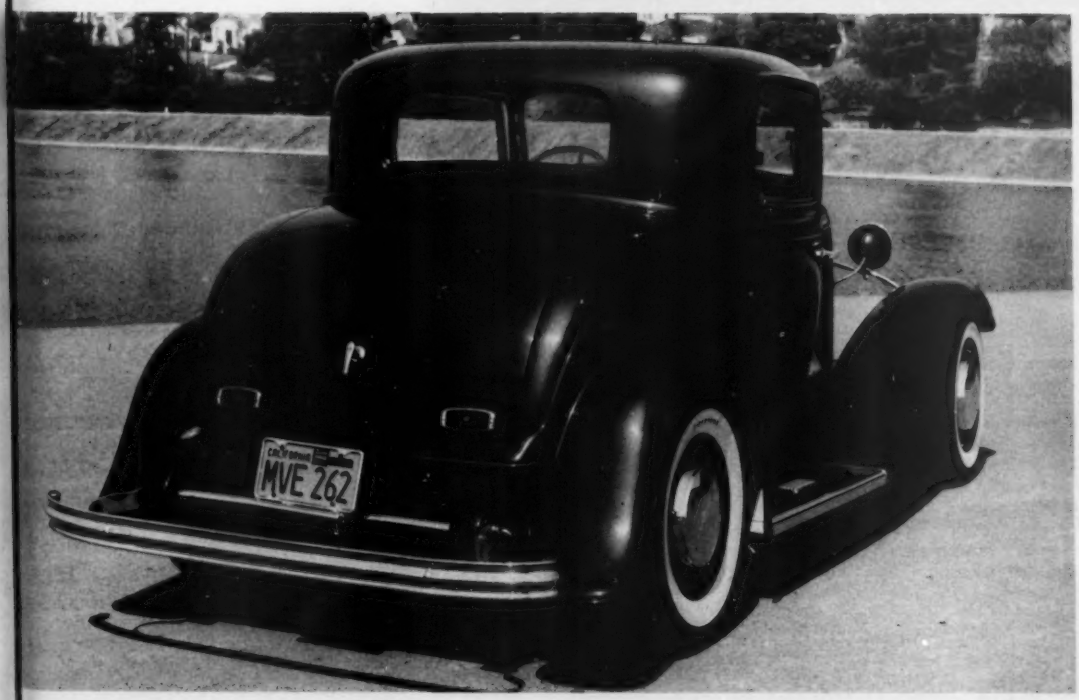
LEFT—Andy Fedora chose to retain the original body lines of his three-window '32 Ford coupe. The body work was done by Jim Burrell of Burbank, Calif. Hubcaps are Mercury. '32 grille shell was filled.

LOWER LEFT—Installation of the near stock '56 Merc engine was handled by Sparks and Bonney. Fedora chose a combination tubular filled dropped front axle, snubbed it with Houdaille shocks, then added a helper set of tube shocks.

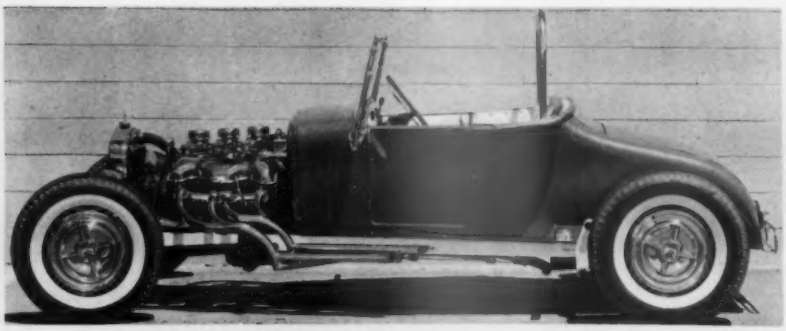
RIGHT—White and grey Naugahyde features thin pleats, diamond motif. Handy fire extinguisher and parking brake handle nestle below bucket seats. Column shift transmission is from a '48 Ford.

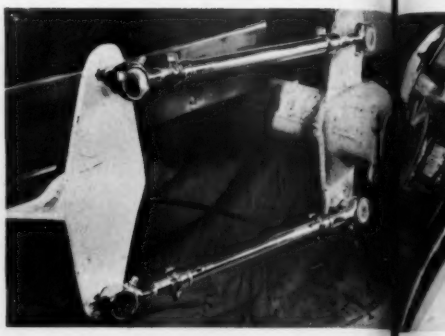
BELOW—Minor departure from original concept is use of Chevy taillights. Bumpers are stock '32 items, paint is Mandarin Red. 6.70 x 15 rear tires are turned by 4.44 gears. Bumpers are '32.

PHOTOS BY BILL ROLLAND



A MOST UNUSUAL 'T'

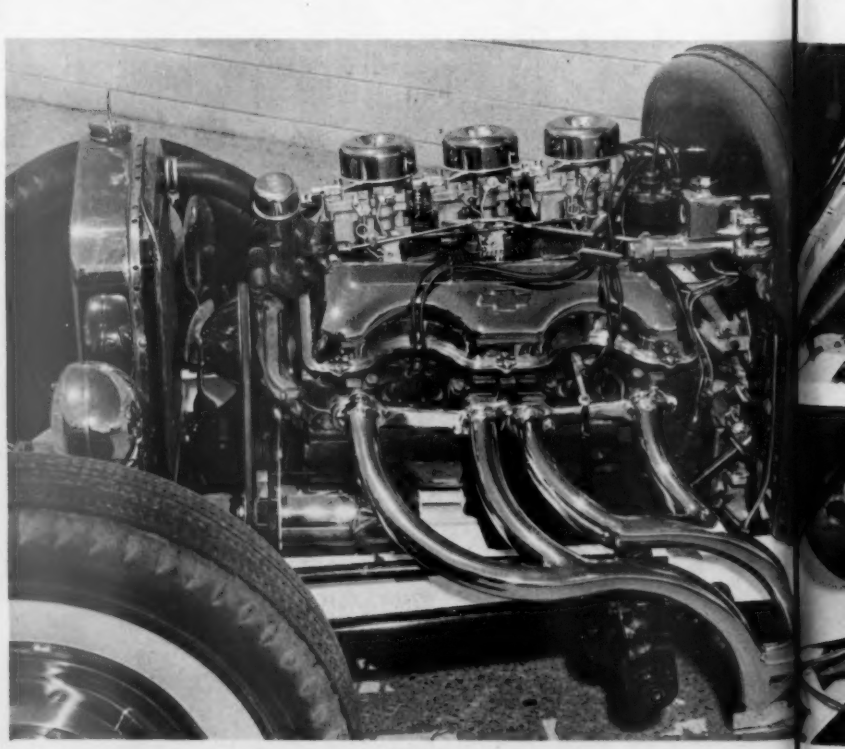




ABOVE—Here are the construction details of the Watts linkage which makes the use of the '58 Chevy axle practical. It has been built from Chevrolet tie rods and eliminates rear axle windup during acceleration. Suspension improvement ideas such as this are not seen often enough on roadster or other early models.

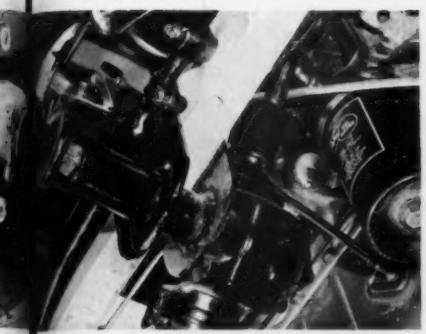
LEFT—Extensive repair was necessary on the '27 'T' roadster body because of rust damage. Channel iron was welded onto sills to add stiffness and to form a body frame before bolting to chassis frame.

RIGHT—Unusual appearing headers were hand formed from electrical conduit. The '58 Chevy engine is the 348 cubic inch model, internally stock with three Rochester carburetors feeding the fuel. For better cooling a special thick-core radiator was built to fit the 'T' shell.

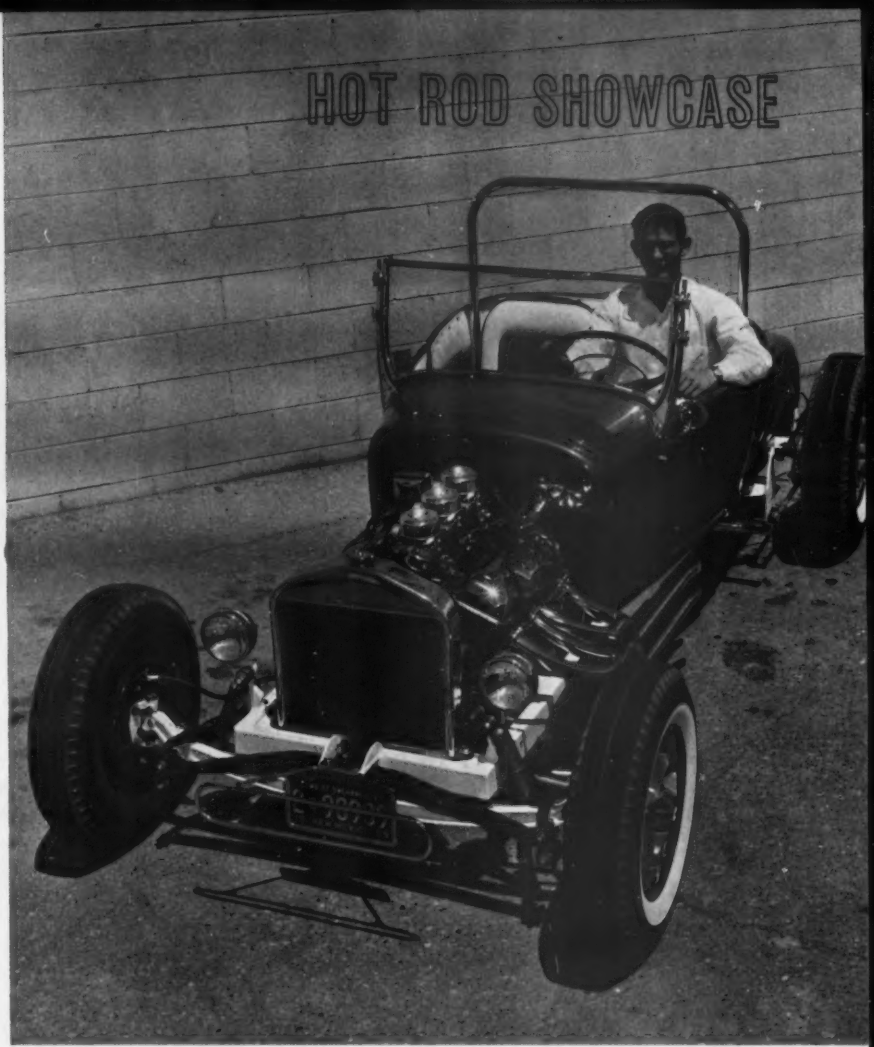


OPPOSITE PAGE—Generally immaculate underside of car pays off in extra show trophies. Frame side members are three-inch channel boxed with ¼-inch plate. Notice in lower photo how the spring perch has been moved slightly back, also the double traction bar setup which helps form a Watts linkage to control torque.

PHOTOS BY HARRY KINNEY

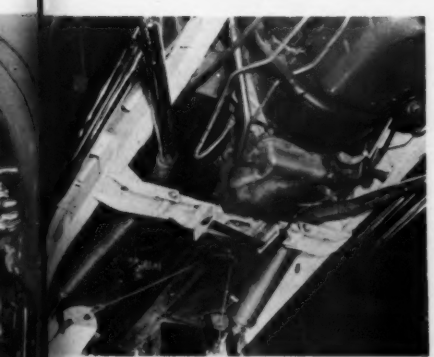


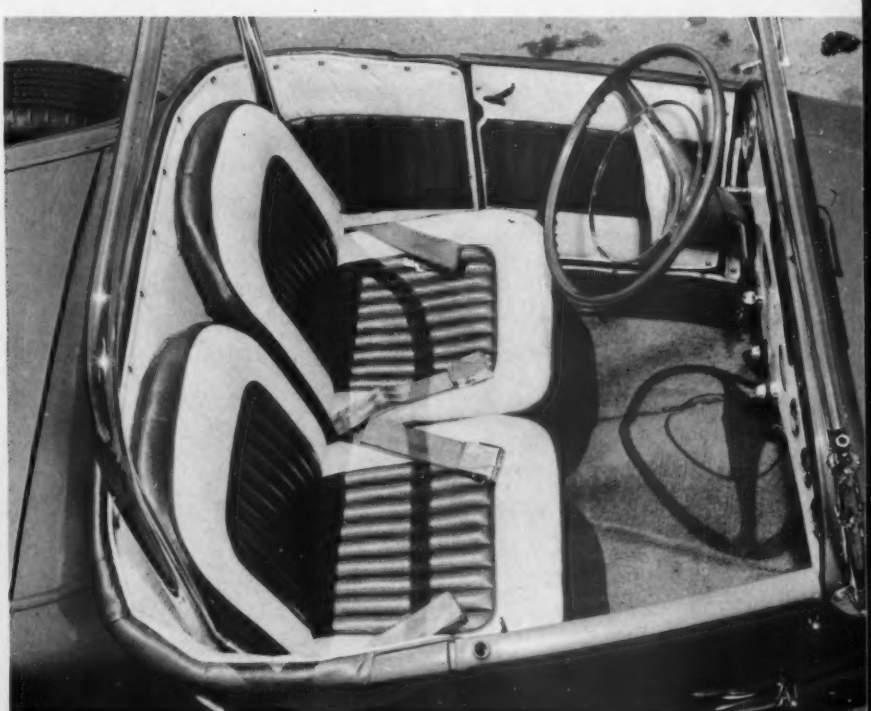
ABOVE—The 1955 Chevrolet steering box has been repositioned to place the drag link up along the frame rail and out of the way. The wishbone, shown at the top of the photo, has been split.



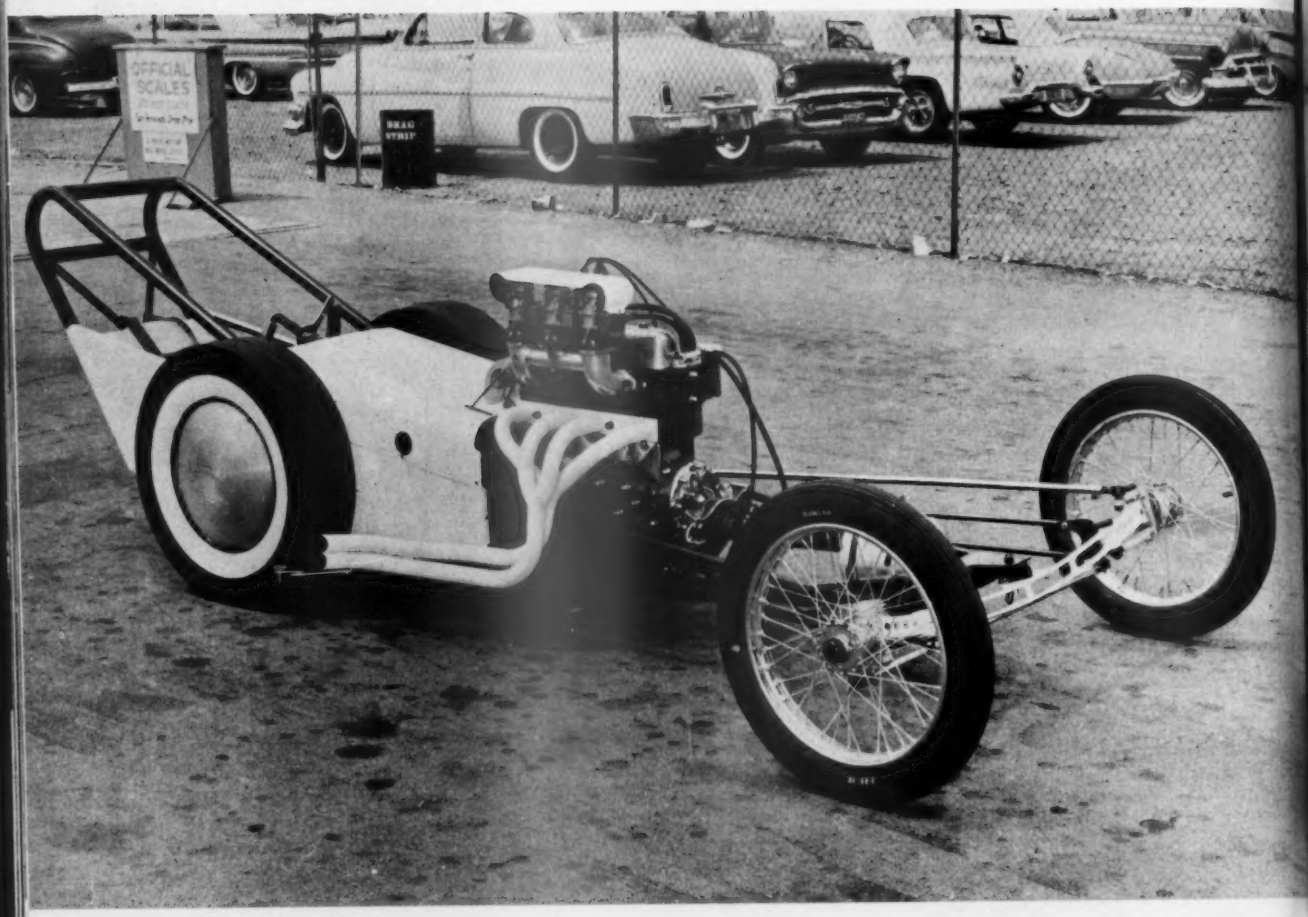
RIGHT—Front view shows more of the frame construction. The front member is 1½ x 3-inch rectangular tube. Axle and spring are '42 Ford, shocks Monroe. Owner Keith Heiskell is behind wheel.

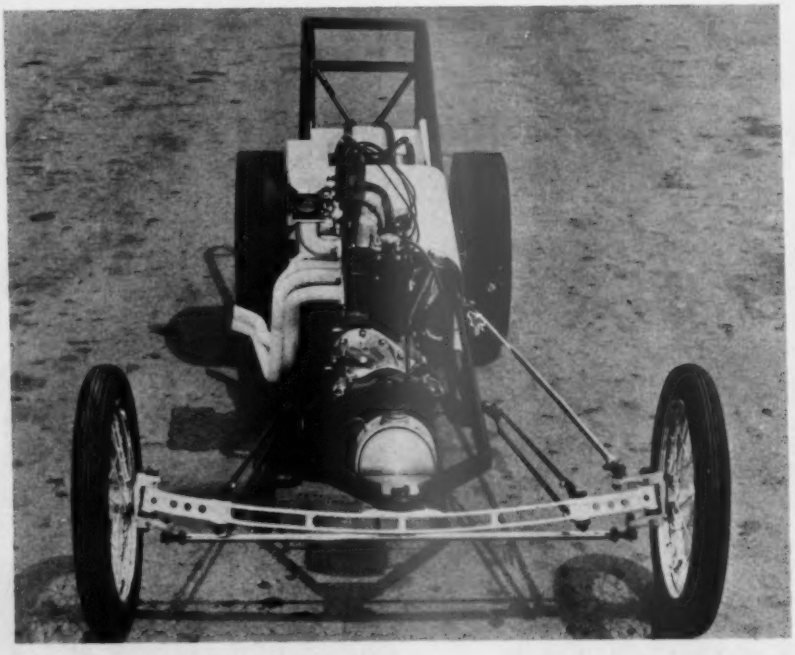
BELOW RIGHT—Seats were custom made, are upholstered in blue and white leatherette. A '58 Chevy wheel and shortened column are used along with Chevy and S-W instruments in Keith's dash panel.

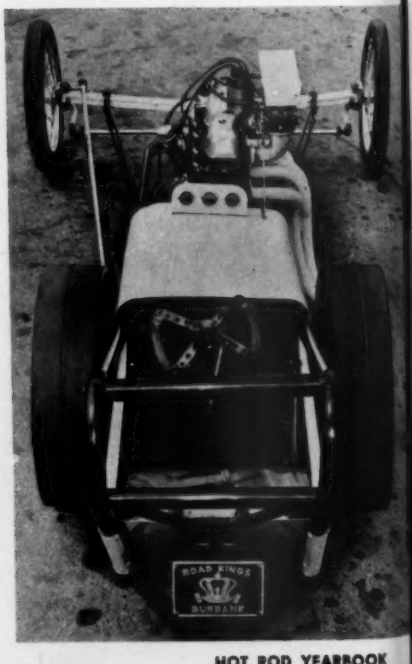




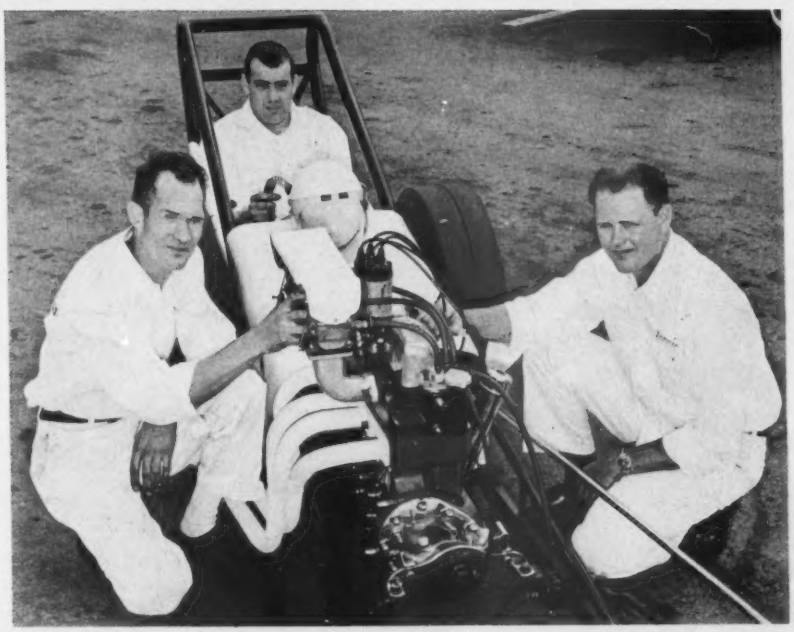
FOUR BANGER







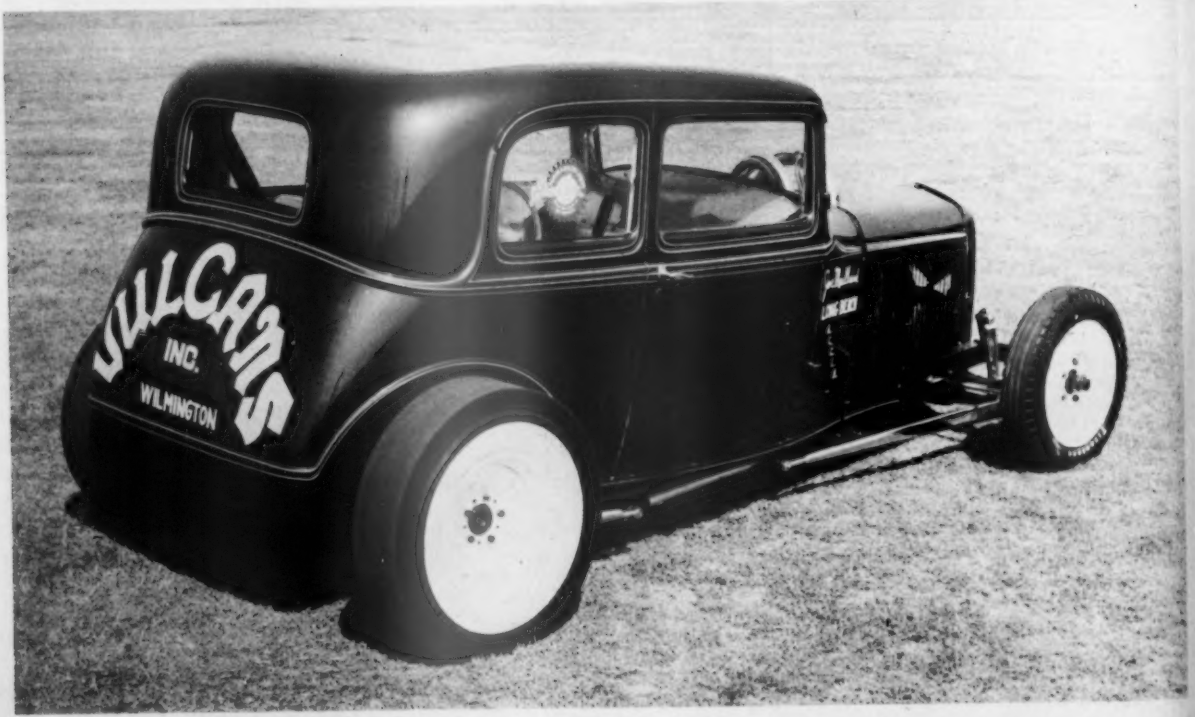
Here's a dragster with a Ford Model B engine! It features a two-port Riley head with three 97 carbs. Goodies on this mill make it turn a total of 150 horsepower. The Model A front axle has been somewhat modified and ventilated. It joins the frame at the center with a rubber compression shock. Spindles are adapted from '46 Ford; steering first graced 1937 Willys. Twisting the Goodyear slicks is a Studebaker rear end with 4.5:1 ratio. Car also features '32 Ford transmission with low, reverse removed. Clutch has Schiefer pressure plate with an "A" truck disc.

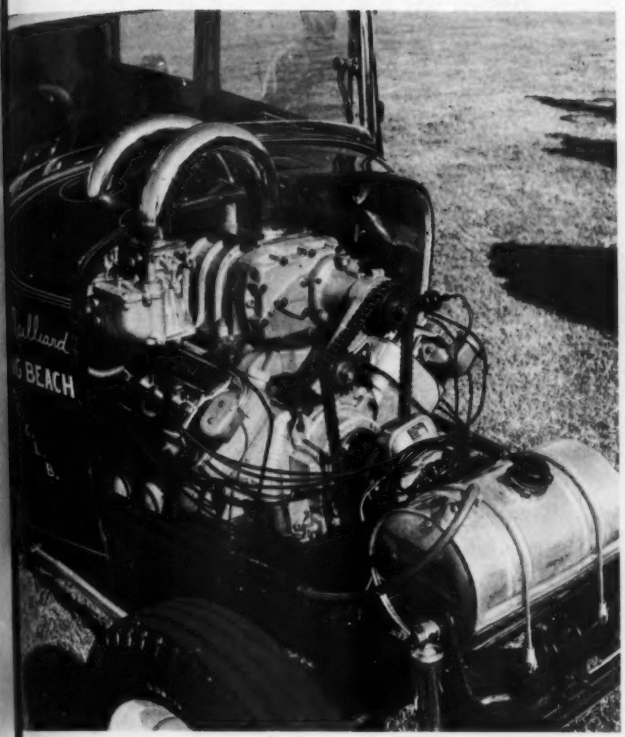


Three man crew, right, includes driver Kent Baber, Owen Engle (left) and unnamed friend. Building car took six months and a low $500 total cash outlay.

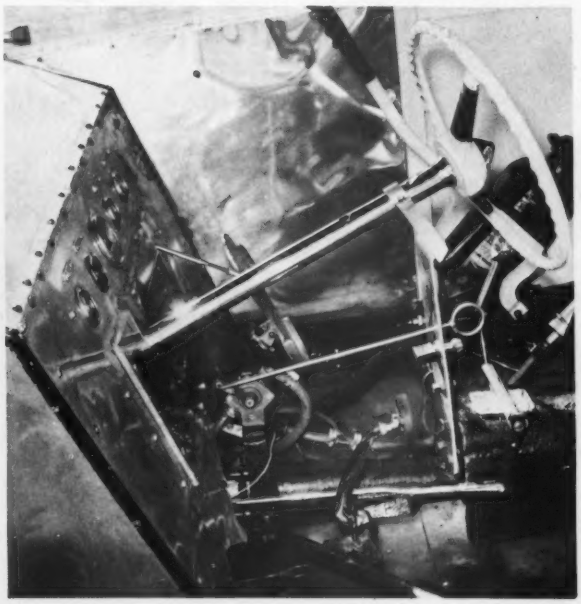
PHOTOS BY PHO CAR

VULCAN'S VICKY





The Vulcans of Wilmington, Calif., have hot rodded something quite unusual, a '32 Ford Victoria. Note the induction tubes arching over the hood. The 1951 Chrysler engine is bored to 338 inches and puts out 450 hp. Unusual updraft carburetors are from surplus landing craft, and feed to a 4-71 blower with Cragar adapter kit. Backing up the engine is Schiefer flywheel, a Hays 3600-pound pressure plate, a '41 Cadillac transmission that uses second, high only, a 4.33:1 locked Pontiac rear end.

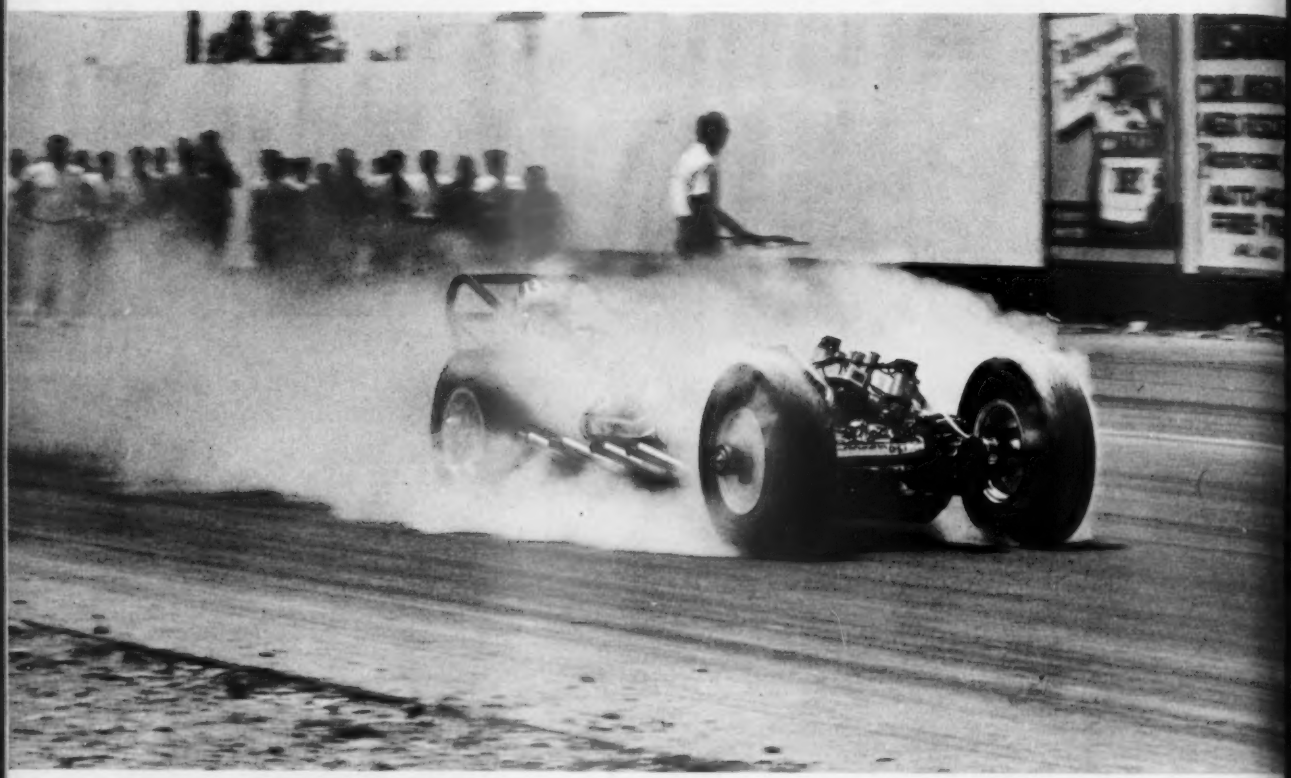


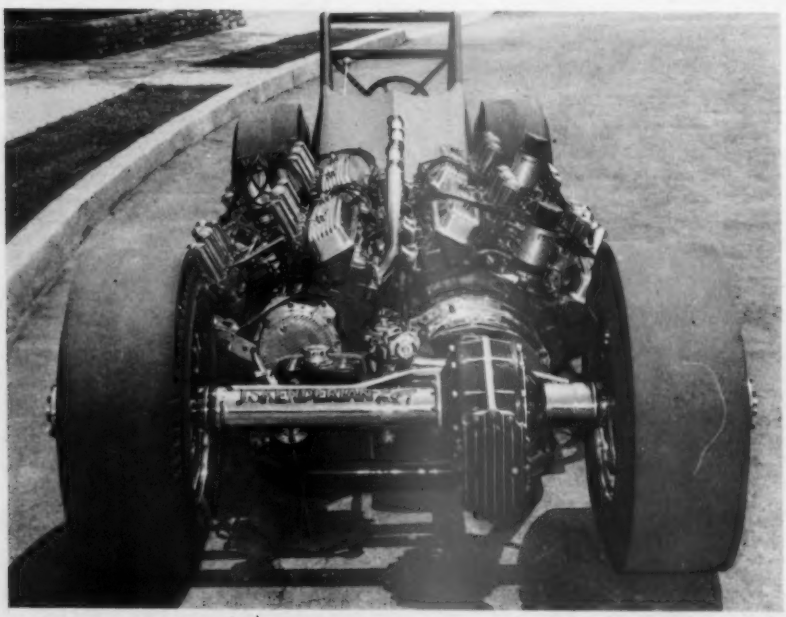
After-end of the cockpit reveals a few more construction details including seat placed at extreme rear of body, ruggedly braced roll bar, long brake handle operating directly on the master cylinder, fuel pressure pump convenient for right hand operation.

LEFT—Front end is simple, with Ford spindles, dropped axle, tubular shocks, '32 Ford steering. Front brakes are not used.

PHOTOS BY CURT HAMILTON

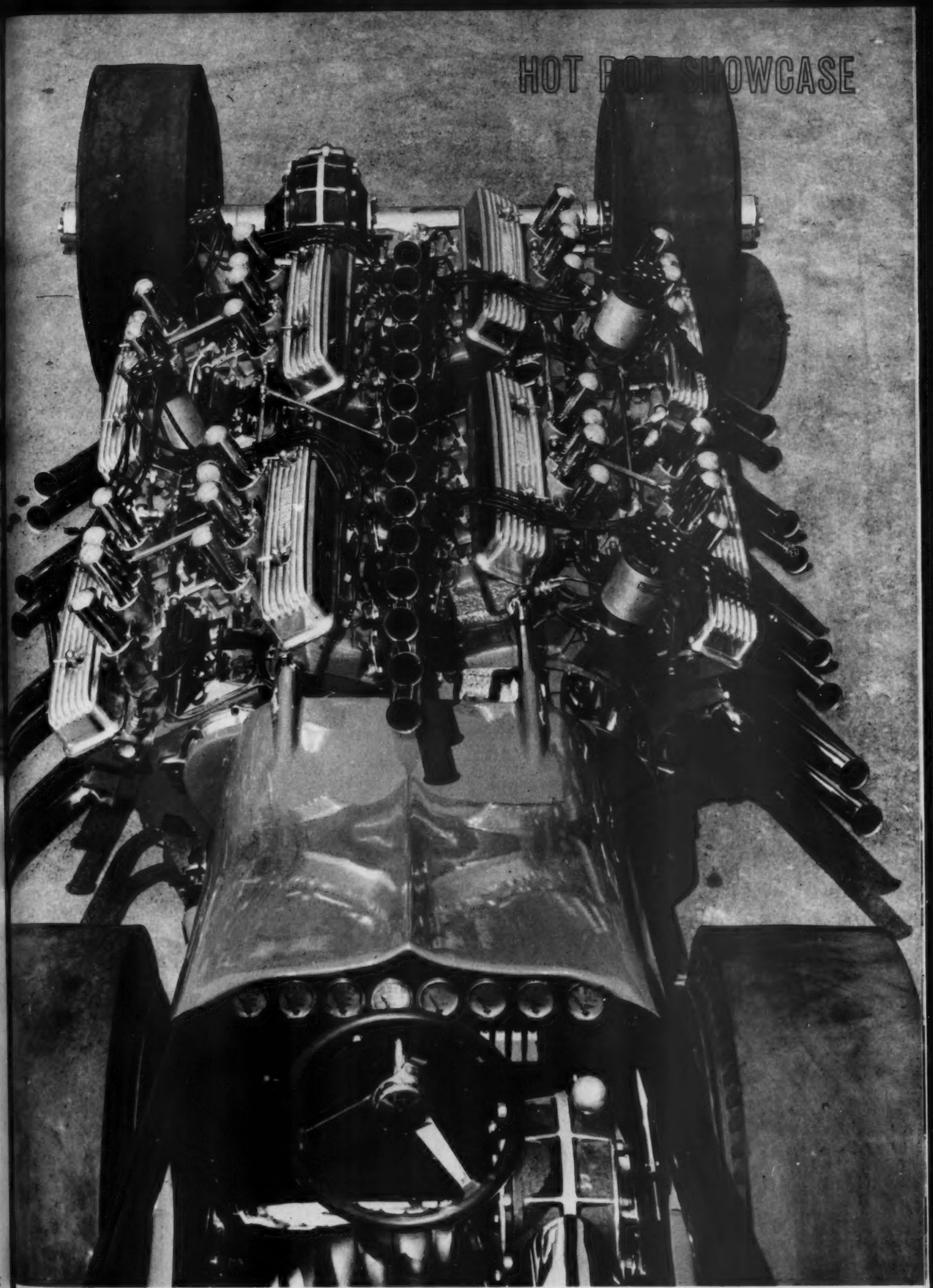
"THIRTY-TWO" CYLINDERS THAT IS!

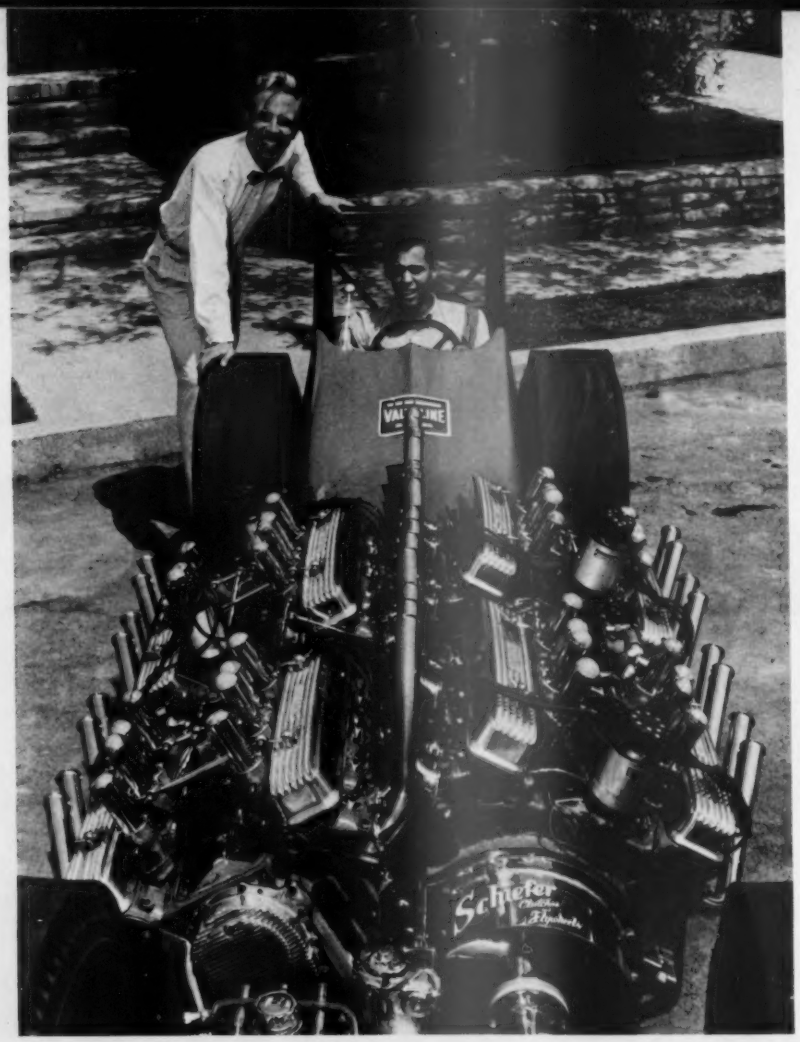




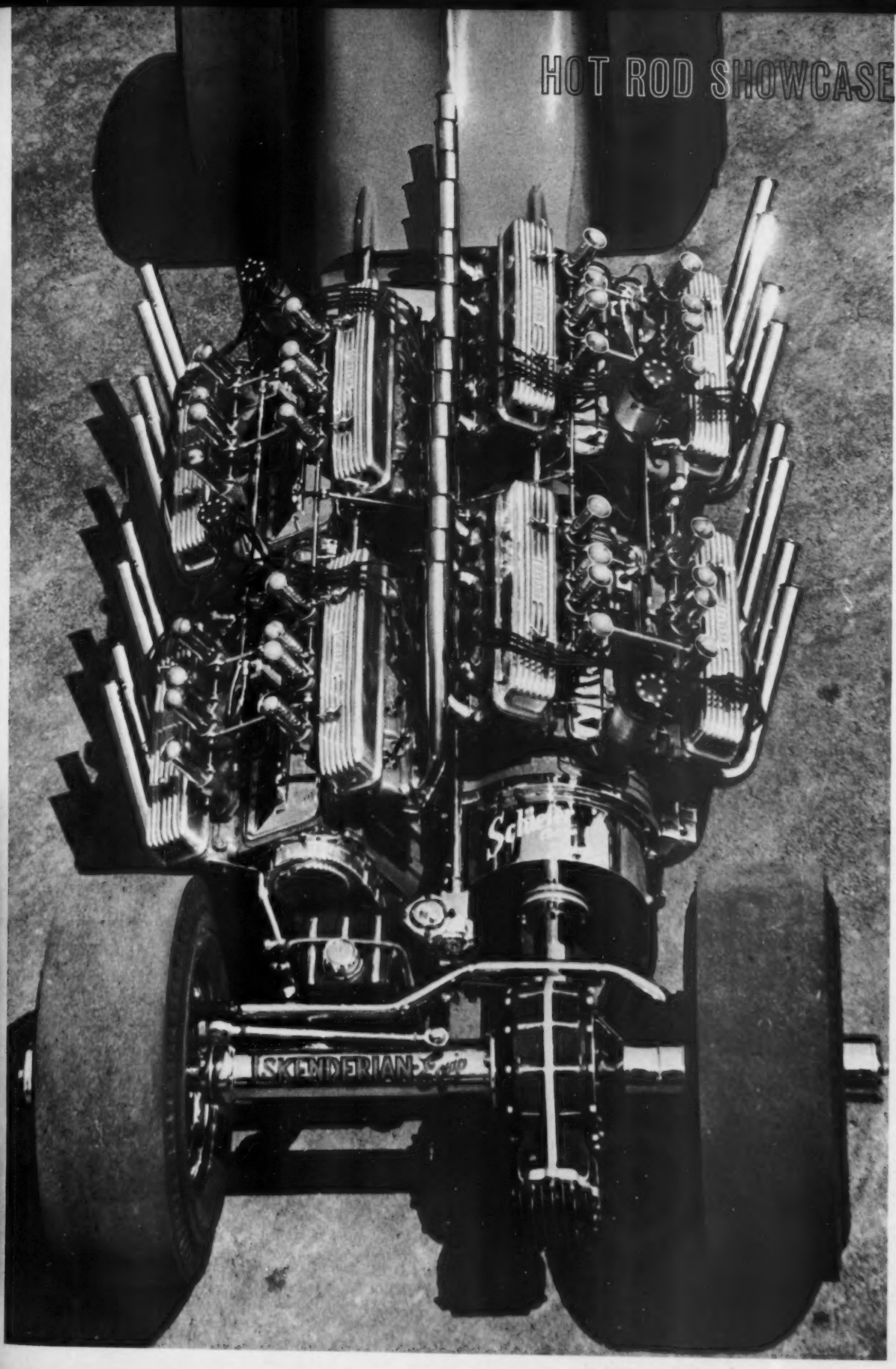
Screaming down strip, Tommy Ivo's four-engine dragster has turned 170 mph, with a 9.29 E.T. Car is powered by four 1961 Buick engines. Two right side mills are coupled via boat-type chain coupler, nose to tail, and drive the rear axle through a Halibrand Champion type quick change rear end. Left side engines rotate in standard direction, but due to their reversed position, rear ends facing forward, their rotation becomes opposite to right bank pair, effectively cancelling out torque effect. They drive forward into another Halibrand quick change rear end that has been modified by switching sides with ring gear to obtain proper front axle rotation. Car total weight: 3555 pounds.

PHOTOS BY ERIC RICKMAN



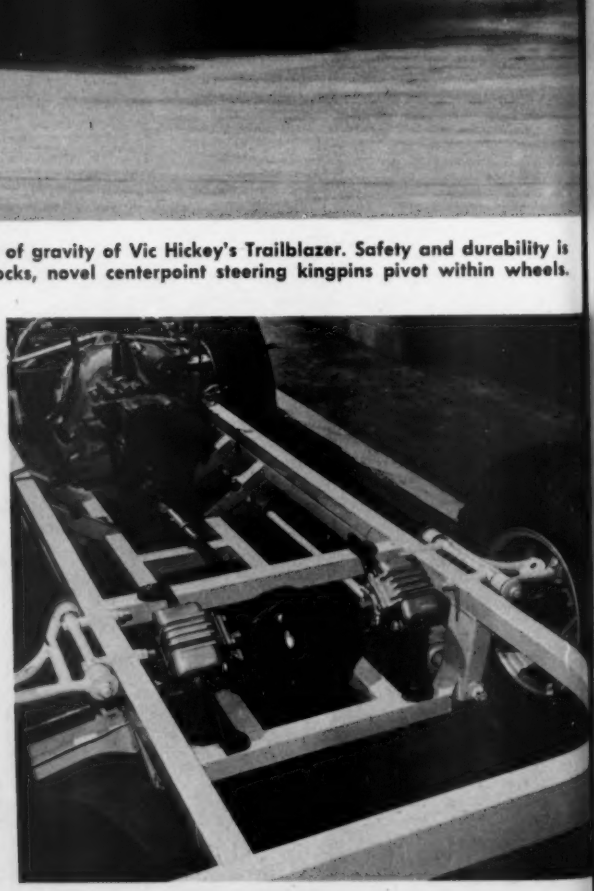


Not including Ivo's time, cost of car at wholesale has been estimated at $12,000. Engines have Iskenderian 5 cycle, flat tappet cams, AC plugs, Hilborn injectors, Scintilla Vertex mags, Schiefer double disc clutch, flywheel assemblies. Rear wheels are Halibrand mag types, fronts being steel fabricated by Ivo, as no wheels were available with sufficient offset to fit over U-joint steering knuckles. Oil pans have been extensively altered for added capacity, plus reshaping due to slant of engines. Left hand pans had to have oil reservoirs and pickups reversed so pumps wouldn't starve under acceleration. Engines carry case and a half of Valvoline oil when the big car is under way.


SKENDERIAN

FWD BLAZER

Broadsliding at high speed is best way to demonstrate low center of gravity of Vic Hickey's Trailblazer. Safety and durability is gained by using Chrysler product torsion bars, double action shocks, novel centerpoint steering kingpins pivot within wheels.



HOT ROD SHOWCASE

Long associated with four-wheel-drive vehicles, and best known for dual wheel adaptors for same, Vic Hickey has finally materialized a dream of several years—the "Trailblazer," a four-wheel-drive roadster for roughing it. Capable of speeds from a crawl to 75 mph, the fiberglass-bodied job covers rugged terrain with all-new approach—in the FWD field—in combining chassis, suspension and powerplant components. Corvair engine is air-cooled; suspension is independent at all four wheels by nearly identical A-arms; torsion bars, tucked safely along frame, are used instead of bulky coil or leaf springs; four inboard Airheart disc brakes are bolted securely to chassis. Working members bolt up to simple two-inch square tube frame of .125 wall to effect an agile, high-stepping machine, as attested by accompanying photographs.

PHOTOS BY ERIC RICKMAN

A FORD FOR FATHER

LEFT- at th his f A. L. been on th abou built work gear brak whic cate

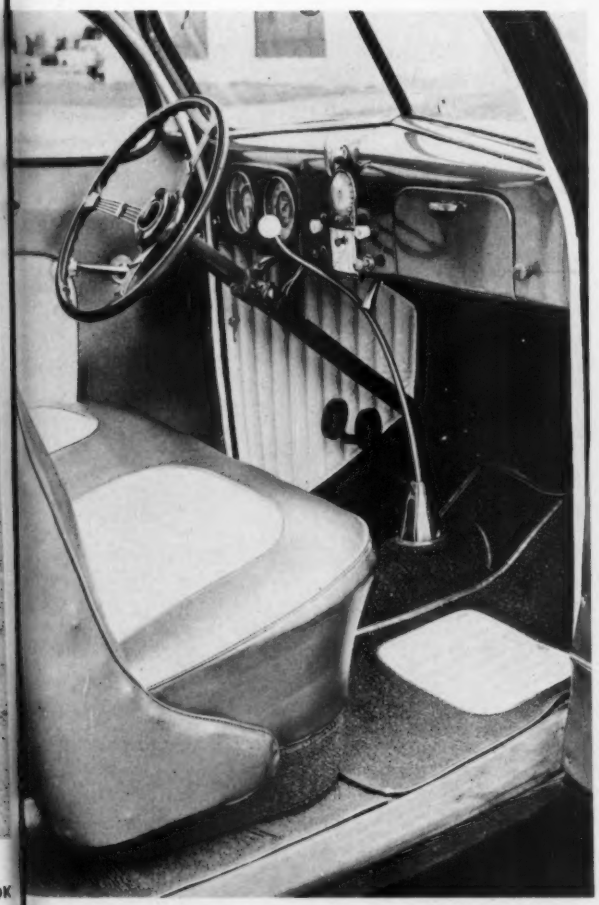
RIGH plete stall and long

HOT ROD SHOWCASE

LEFT—Harold Hyde liked to run his cars at the drags and occasionally recruited his father's mechanical assistance. When A. L. Hyde discovered a '37 Ford that had been in storage for years with 3000 miles on the odometer, he decided that it was about time to reverse the procedure; he built it up for himself but put Harold to work as an assistant. The entire running gear is stock '37 even to mechanical brakes. From the rear, only dual exhausts which tie into headers on the engine indicate that everything up front is not stock.

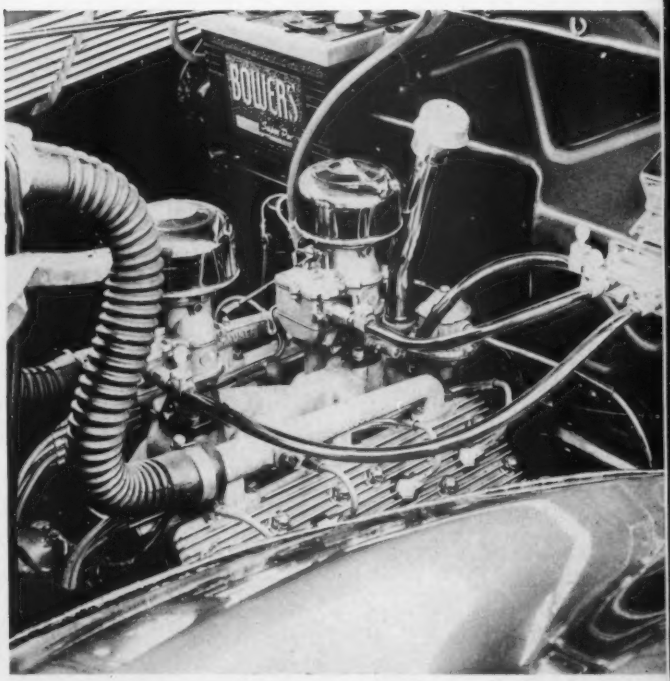
RIGHT—The only deviation from a completely stock '37 exterior was the installation of sealed beam headlights and a bright red paint job. After the long storage, all the chrome was redone.

PHOTOS BY BOB HEGGE



LEFT—Interior is mostly stock but fitted with modern rolled and pleated leather seats and carpeted floor. Skelton Trim Shop of Atlanta, Georgia, did the special upholstery work.

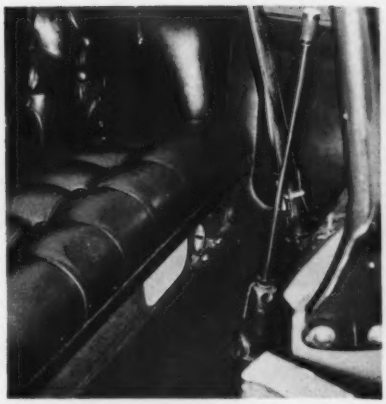
Beneath hood rests rare sight nowadays; Ford 60 hp V8 with .100-inch oversize pistons for whopping 146 cu. in. Winfield cam, Offy heads, Edelbrock dual intake help pep performance.



Authenticity is preserved by use of 'T' gas tank moved to rear, while step plates on running boards were made to match original castings. 'T' steering wheel passes under hand-made dashboard. Note: Model A bumpers, '48 wheels, Mamie Van Doren.

PHOTOS BY ERIC RICKMAN

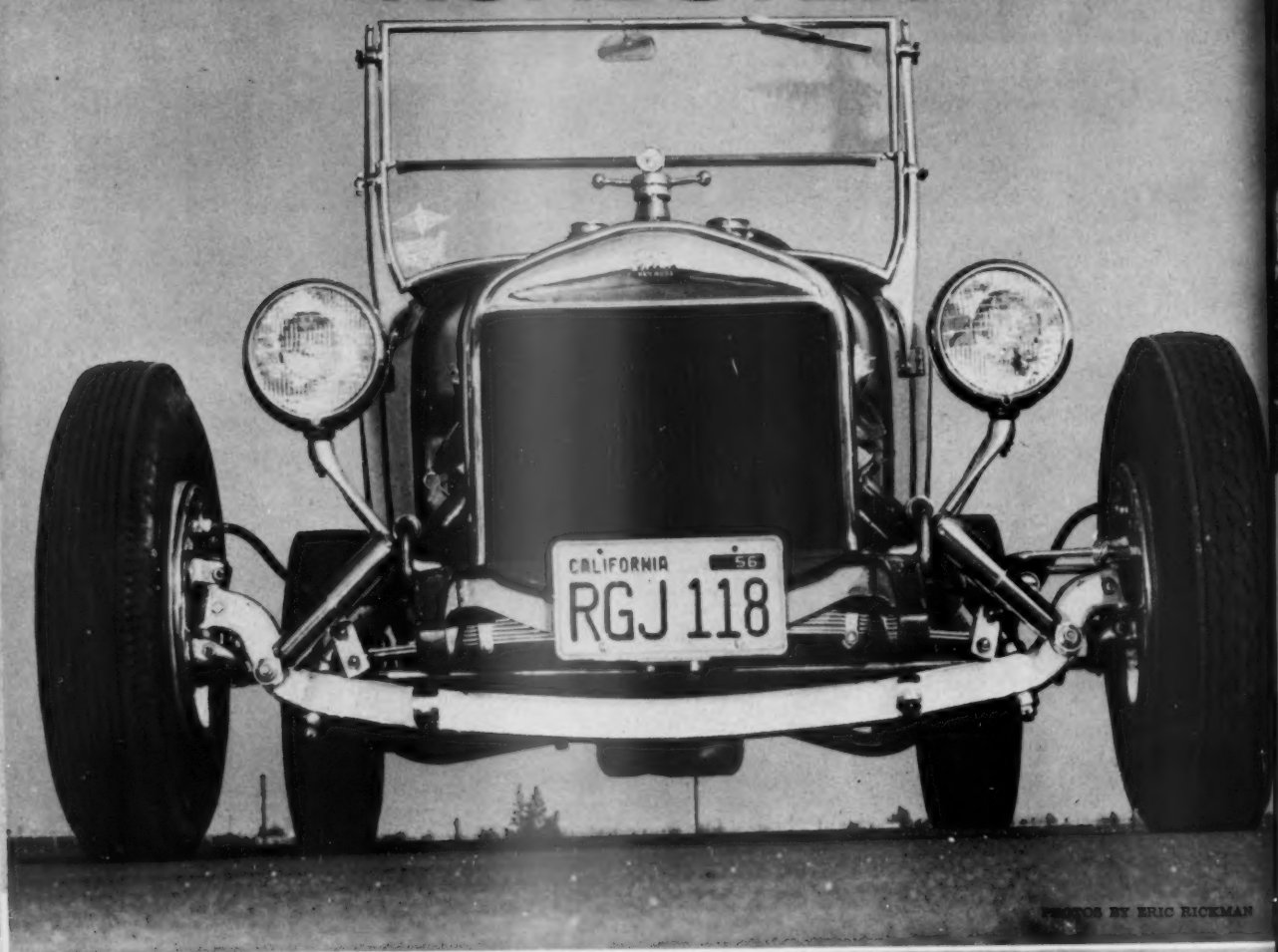
RIGHT—A radio is nice but it's best to conceal it under the seat in this case. Gear lever leads to Ford transmission and 3.78:1 ratio differential. Power is supplied by a '48 Mercury flathead that will shortly be replaced by a Chevy V8. Grabowski reportedly was offered $5000 for car. He refused, understandably so.



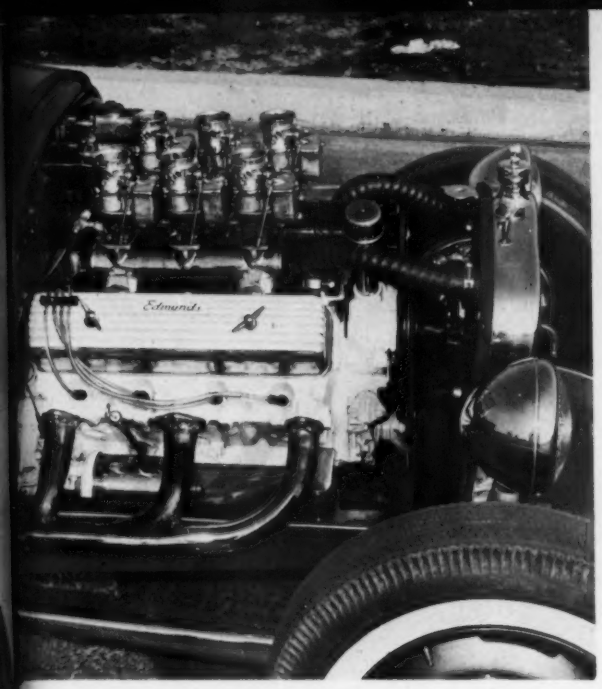
FAR RIGHT—Bill Colgan's Auto Upholstery did deep diamond tufted pleats, a style popular on many expensive cars of this era because it reminded owners of carriage days. Grabowski wants better than buggy handling for his tub, uses '56 Ford pickup steering mounted at angle.

With justifiable pride, Norm Grabowski shows beautiful custom restoration of his '25 'T'. Valley Custom did the body work.

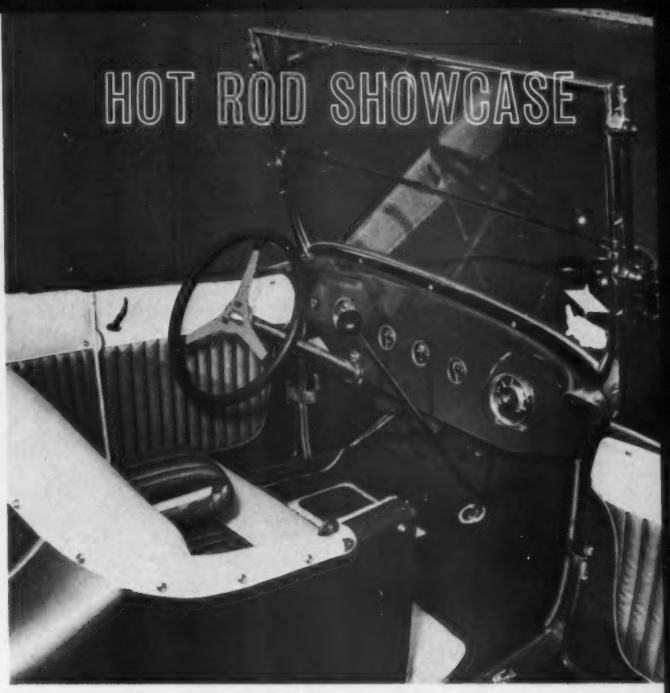
FLAWLESS ROADSTER



PHOTOS BY ERIC RICKMAN



HOT ROD SHOWCASE



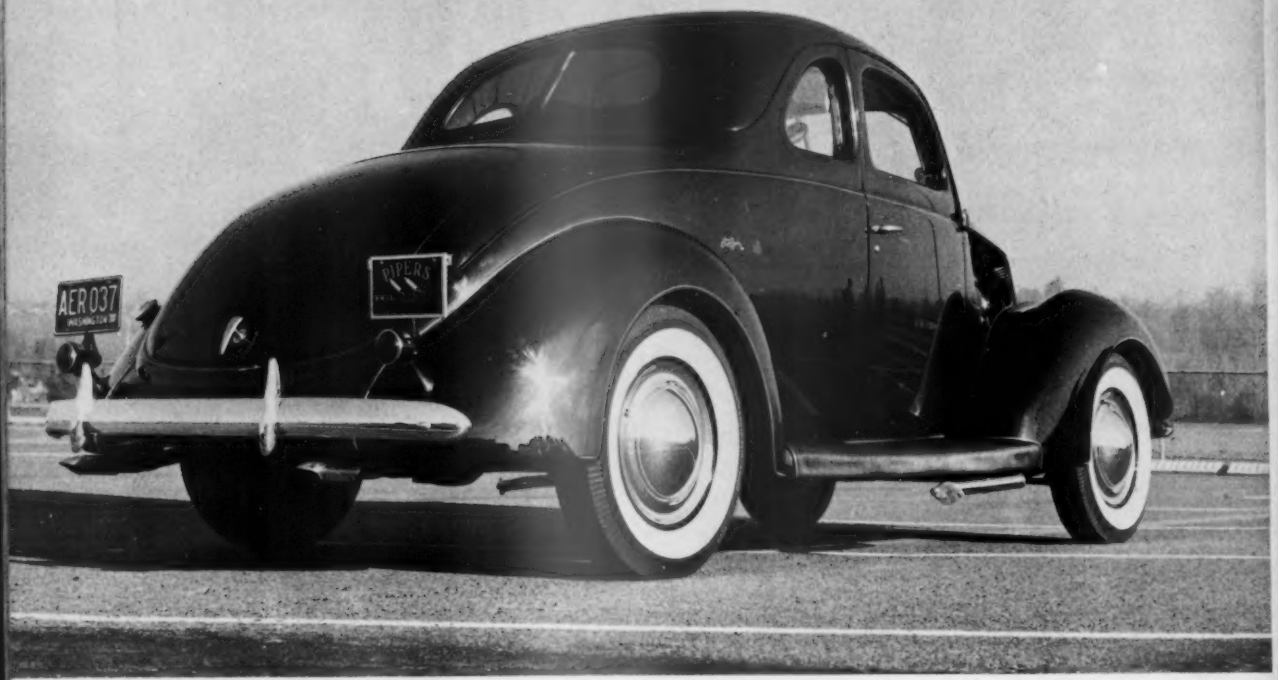
Dale Gould's "flawless roadster" is powered by a 1952 Olds engine displacing 365 cubic inches. It has big valves, an Iskenderian cam and kit, '56 Olds heads and a Crower U-fab intake manifold. Six Stromberg 97 carburetors supply the fuel.

Bell steering wheel is fitted to '37 Plymouth steering gear. Custom dash features the usual array of instruments. Hanging pedals operate linkage from Buick, Chevy parts. Upholstery by Tony Nancy. Long floor shift tops a Cad-La Salle transmission.

Special gas tank has been built and fitted to extreme rear of body. Bruce slicks take thrust through 4.11 differential gears and 18-inch open driveline from a '46 Ford panel truck. The rear end has recently been completely chrome plated.

A member of the L.A. Roadster club, Dale uses the car for daily transportation, yet keeps machine flawless. 12-volt battery, tools and emergency equipment rest in the carpeted trunk. Prior to recent work, Dale had $2000 in his roadster.

'37 REVISITED



The 1937 Ford would rate in the "most often overlooked" category, but Dick Flynn has resurrected his coupe to mint condition. All stock chrome has been replated and a '39 Ford rear end sporting '48 Lincoln brakes is featured. Columbus shocks help the coupe handle properly while Firestone tires work to maintain traction. Car is painted glittering jet black.

Trunk area is carpeted in black, holds safety items. Chrome beading has been installed between body and fender joints, otherwise exterior appearance is perfectly stock, a theme that is becoming increasingly more popular with rodders.

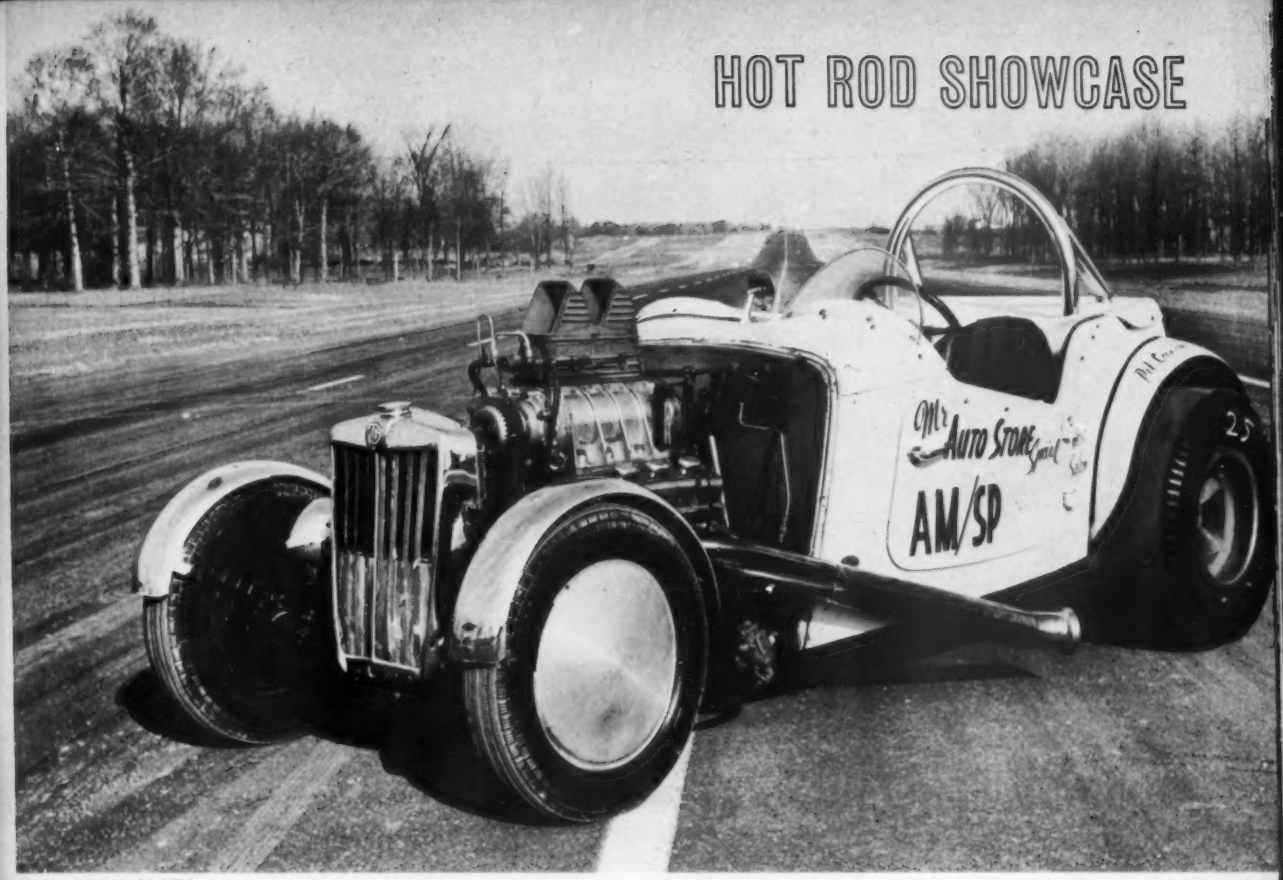
Smitty of Seattle handled the stitching chores that resulted in black and white Naugahyde. A '39 Ford furnished the steering gear as well as the transmission which is adapted to engine via a '50 Merc bellhousing; clutch is '50 Mercury.

ABOVE—The 1949 Ford engine has been modified by a poke and stroke to 280 cubic inches, Offenhauser heads and intake manifold, Fenton exhaust manifolds, full balance and a Mallory ignition. Oversize valves are actuated by a Millings cam and Johnson lifters. The radiator was pirated from 1946 Ford six and adapted.

RIGHT—Dick Flynn built four hot rods before the present one, feels that the '37 is best of all his rods. Dick bought car for spare parts but decided to build it up when he couldn't sell what he did not need. The '37 front axle carries '48 spindles and 1948 Lincoln brakes.

PHOTOS BY PETE SUKALAC

SPORTS ROD



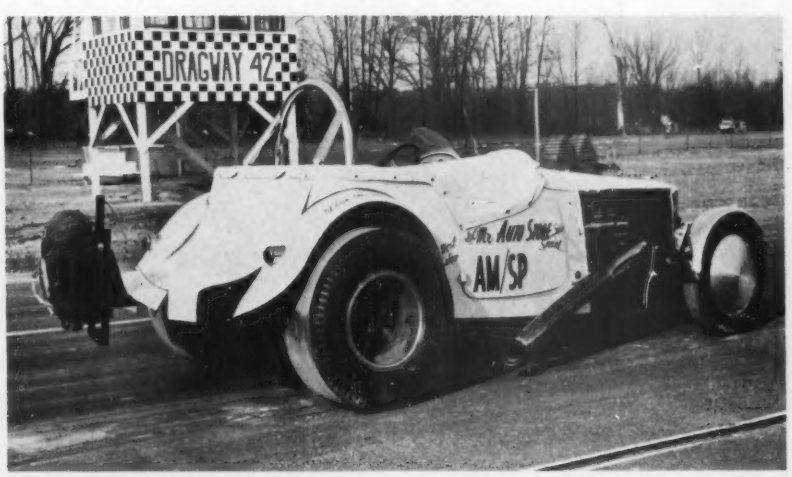
PHOTOS BY BOB DANIELS

Left—Dave Billman has owned six drag racing cars, all supercharged Chevy powered but one, and all were record holders. Present machine holds A Modified Sport top speed record, 135.95 mph. Dave owns a speed shop and smoke off rear wheels indicates he knows his parts.

Above—No matter how you look at it, this is a hot rod. Anglia supplied the front axle assembly, which presents 50 inches tread width. Front wheels are of Ford make, hold Firestone tires which are covered by cycle-type fenders to keep the car legal for class. Grille is MG.

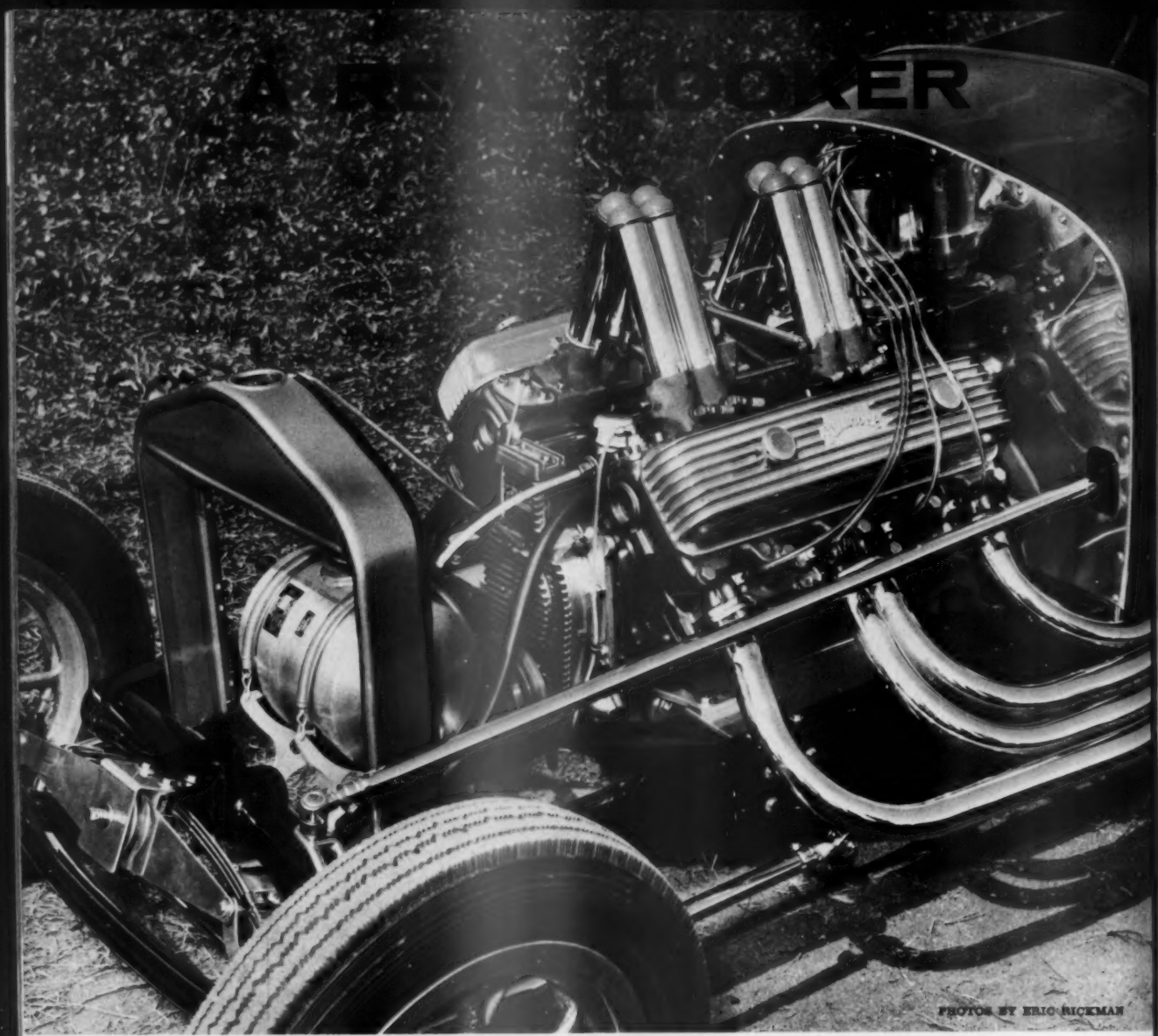
Below—The quarter-mile traction is handled by M and H slicks mounted on American mag wheels turned by a 4.56 ring and pinion in a '59 Chevy rear axle. Dave has found that drag chute is nice safety item, helps slow '53 MG very nicely. Exhaust system is homemade.

Far Left—The cliche, "What's up front that counts" is certainly true in this case. The '59 Chevy displaces 340 cubic inches, features a Giavannoni roller cam and Forgedtrue blower pistons, truck heads, Edelbrock manifold, Enderle injector, Vertex mag, Ansen scattershield-bell housing and a Schiefer clutch assembly. The big Chevy is blown with 6-71 GMC.

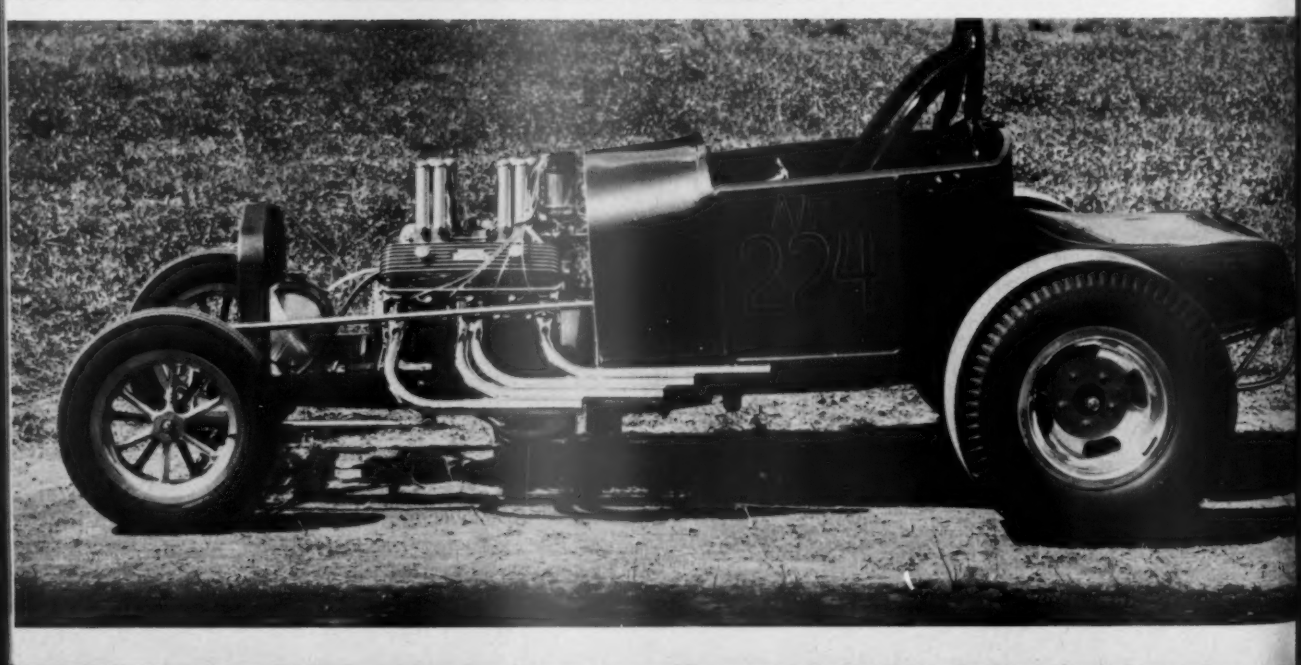


Left—Original wood dash panel has been replaced with aluminum unit holding necessary instruments to record blown Chevrolet engine's characteristics. A 1948 Crosley furnished steering control which originates in a cockpit covered in black leather by Marsh of Mansfield, Ohio. Note shoulder harness.

A REAL LOOKER



PHOTOS BY ERIC RICKMAN

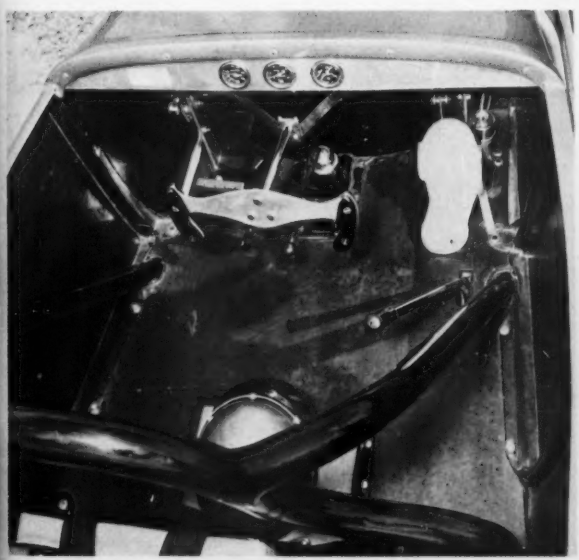


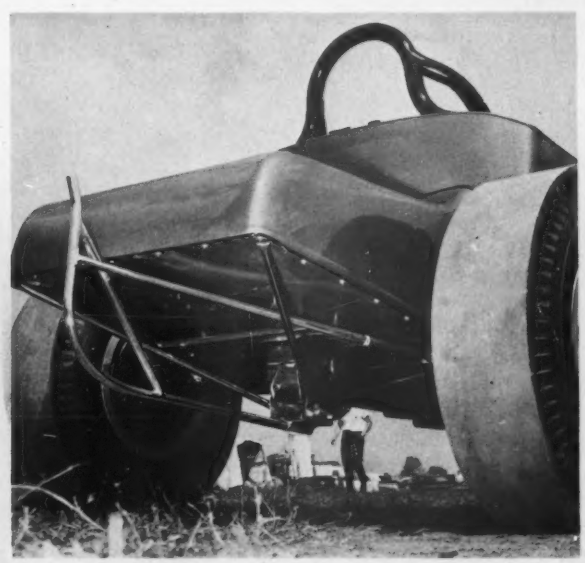
C. J. Kearney's drag roadster features a 1948 Anglia beam axle to spread front tires to 47½-inch tread width. Friction shocks are homemade, as is the unusually long steering arm that mounts to the Anglia spindles. 'T' radiator shell is required by rules, adds distinction to rod.

LEFT—Home for the horses of Kearney's machine is this 461 cubic inch '57 Olds built by friend LeBoeuf. Its power is gained through an Iskenderian cam and kit, Forgedtrue pistons with Grant rings, milled heads giving 13:1 c.r., mag modified by Joe Hunt, and Hilborn injectors. Front cover plate is by Moon Equipment.

BELOW RIGHT—Attention to minute details is everywhere on Kearney car. The turtle deck floor has been replaced with aluminum sheeting fitted perfectly. Supporting braces for deck run from Ford rear end holding 3.78 ring and pinion. '48 Lincoln brakes have backing plates drilled in exacting pattern for cooling.

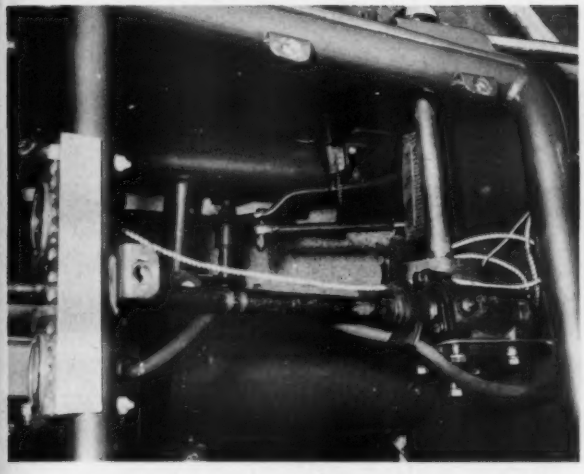
Complete interior of roadster has been covered with blue and black Naugahyde by Cannon Brothers of New Orleans. Steering wheel transmits to front through Crosley box modified for center mounting.

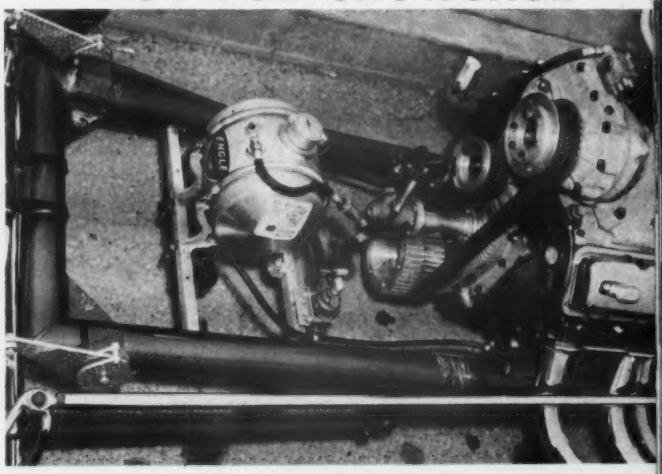




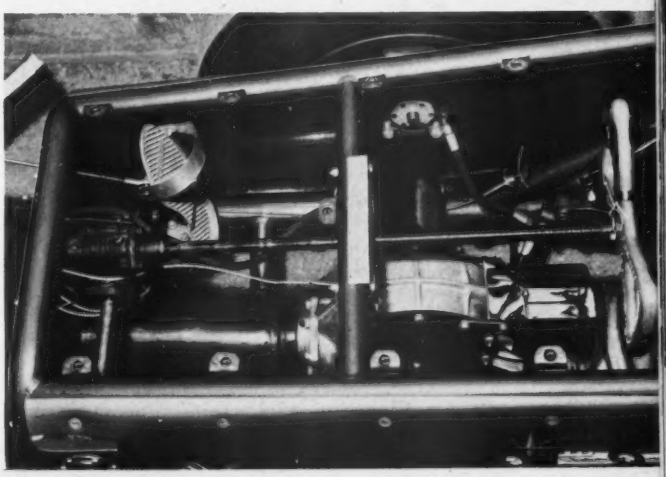
THE ELIMINATOR

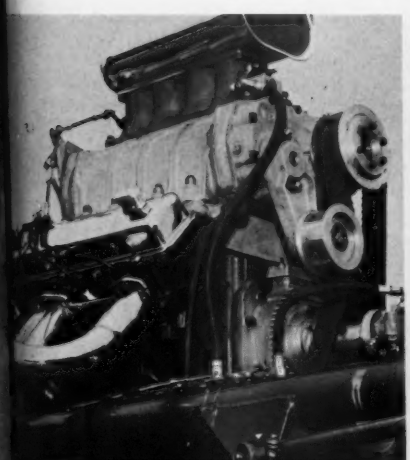
PHOTOS BY ERIC RICKMAN, BOB D'OLIVO



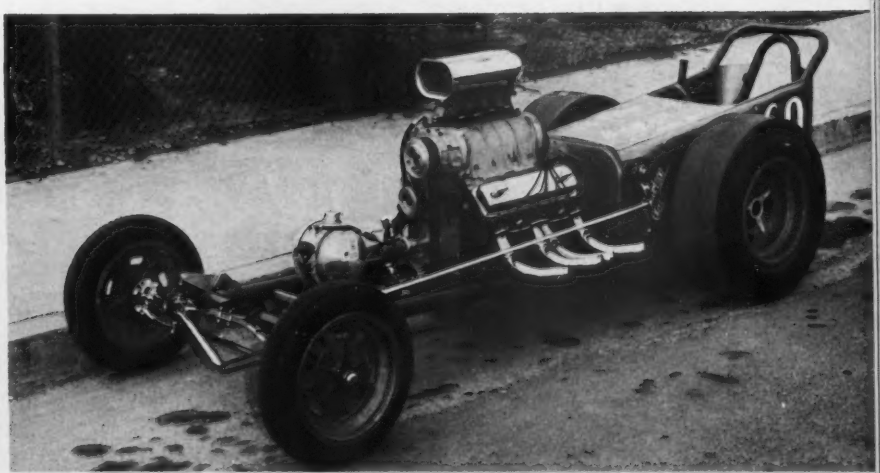


1960 Nationals winner is loaded with little features that make it outstanding. For instance, when car is being tuned while at a standstill, rod is slipped above depressed clutch arm, eliminating possibility of an accidental engagement of the clutch. Note cover over the short driveline. Water overflow catch can is mounted just below aluminum gas tank. Fuel passes through filter, shut-off valve before entering injector metering box. Car has turned over 170 mph in the 8 second bracket since the National drags. Heim fittings are mounted on the ends of a long drag link. Removal of top body panel reveals driving office. Crosley steering shaft is supported by center brace, loop on foot pedal ensures positive release of throttle. Rear brakes are Halibrand-spot items while quick-change rear end is also of Halibrand manufacture, running a 3.78 gear ratio. Car sponsor, Albertson Olds, Santa Monica.





Powerplant of the 1960 Eliminator car is a 1959 Oldsmobile having a square bore and stroke at 4⅜ inches, displaces a total of 462 cubic inches. An Engle cam, push rods, and lifters team with Gotha rocker arms to help mill breathe. Pistons are Forgedtrue of 8 to 1 compression ratio with Grant piston rings.



Albertson Olds rail displays brutish good looks, partially due to rather forward location of engine, knock-off rear wheels by Halibrand. Ronald Scrima is responsible for the chassis while Gene Adams fostered the big engine. Scrima is widely known in drag racing circles, as is Adams who ran screaming Olds powered A Gas coupe in past. Important chassis dimensions: wheelbase, 93 inches; front tread width, 46 inches; rear tread width, 32 inches. Extreme king pin inclination helps front end maintain alignment when weight shifts to rear during acceleration. Car turned e.t. of 9.25 seconds at Nationals. 6-71 GMC blower driven by Gilmer timing belt.

NHRA—INGENUITY IN ACTION

ORGANIZED hot rodding, on a nation-wide basis, came into being in 1951 when the National Hot Rod Association was formed. Established as a non-profit corporation in California, NHRA's primary objective was to help bind car clubs closer together and thus enable them to elevate their standing in the community. By providing a guide pattern, sample constitution and a basic program for activities, NHRA successfully inspired the formation of countless new car clubs from coast to coast, and it wasn't long before affiliates were established in many foreign lands. By encouraging clubs to participate in their local civic programs and various beneficial campaigns, NHRA effectively steered the entire hot rod movement into a higher level of operation than it had previously known, and enforcement and civic leaders were quick to recognize the new look that organized hot rodding was taking on. Although individual memberships were welcomed, the main motivation was with the expansion of the car clubs movement, a keystone on which the sport has been built.

At the same time, it was evident to the founders of NHRA (all of whom were active hot rodders with a background of dry lakes and Bonneville speed trials experience) that there was a crying need for some suitable competition medium in which the members far and wide could participate. Speed trials facilities were limited almost exclusively to the California dry lake beds, which were becoming rougher each year, and the once-a-year activity at the Bonneville salt flats. The problem was to establish a new outlet for performance that could be conducted with safety, and one whose essential facilities could be obtained at or near any community.

Quarter-mile drag racing had been practiced in a few locations on the West Coast, usually on some abandoned or little used airfield runway. No specific rules had been set, no standard procedures, nor was there any practical insurance coverage available to protect the property owners, much less those who instigated and took part in the meets. All of these needs gave NHRA a ready entree into the field of drag racing, and it was only a short time until classes, safety rules, operating procedures and even a specialized insurance program materialized which afforded ample liability protection for sponsors and land-owners, plus introducing the first participants insurance coverage program for drag racing.

Thus came about the National Hot Rod Association's full-fledged sanction program for drag strip operators, recognized as the one big step that helped put drag racing on the map as a legitimate major sport. By providing the guide, producing the rules, and actively assisting with the establishment of new strip facilities in all parts of the country, NHRA representatives laid the foundation upon which today's successful drag racing enterprises are conducted—the motto, Dedicated To Safety.

A year after its drag sanctions program was launched,

One of NHRA's early affiliate groups was Piston Pushers Club, Rochester, N.Y., shown in one of the weekend club events.

One of the most active early NHRA charter clubs was Redwood Kings, Eureka, Calif., shown at NHRA booth, Oakland show.

Above—NHRA's first Safety Safari made tour of U.S. in '54, spreading gospel of safe rodding, helping to unify the sport.

Right—The 1st Regional Championship drag race conducted as part of the Safety Safari tour was at Caddo Mills, Texas.

Below—Typical early groups that assisted NHRA in developing drag racing as a safe, respected sport were Louisiana officials, representing the State High Patrol, Caddo-Bossier Safety Council, DeSoto Parish Sheriff's Office, and Shreveport P. D.

NHRA formed the forerunner of its well known "Safety Safaris," in which a select crew of drag racing experts was sent on tour of the United States to help with the coordination of the new sport in all areas. Their principal objective was to teach the application of safety to those who conducted the events as well as the ones who took part in the competition. The beneficial effects of the Safari's visits were immediately evident in the improved nature of the vehicles themselves and in the manner with which events were conducted after the Safety Safari had been in each locale visited.

To add interest and color to the Safety Safari, NHRA introduced a series of Regional Championship drag events at suitable sites along their route, and it was at these highly successful events that the most effective accomplishments were made. The regionals provided a popular gathering place where hot rodders could gain first-hand information on the best known ways with which to enjoy their sport, with competition at its keenest. As an added finale to each year's Safari tours, NHRA established the big National Championship Drag Races, conducted under supervision of the Safari crew and NHRA regional advisors from all parts of the U. S. Right from the first, the NHRA Nationals became hot rodding's number-one event, and the annual BIG GO, as it is also called, has grown continuously ever since. The Safety Safari was on tour in 1954, 1955 and 1956, then NHRA expanded its field advisory system and multiplied the Safari services by having representatives at hand in most parts of the country who could assist and advise in NHRA's behalf, thus helping to further develop the sport in all areas.

Always seeking to improve its program for organized drag racing, NHRA later assigned full-time Division Directors to the task of overseeing the activities in their respective territories. Working with the Division Directors were Regional and Area Advisors, all of whom are volunteers, in a vast network that encompasses all of the U. S. and Canada. Through the hard-working contributions of these dedicated people, NHRA has become the trail blazer, the pioneer and the custodian of the drag racing world.

Proof of the effectiveness of NHRA's drag racing program is readily seen in the volume of participation in its annual sanctions. Within a single typical year, NHRA sanctions well over a thousand events at almost 100 strips. A season's car entries will total more than 170,000, while spectators mount up to almost two million in all. Considering the number of runs each car makes through the course during each of these events, vs. an amazingly low accident experience, the safety record of drag racing is among the highest in the entire realm of sports activities.

As to the National Championship events, the first of these was presented by NHRA at Great Bend, Kansas, in 1955. The second took place at Kansas City, Missouri, then the third and fourth were held at Oklahoma City, Oklahoma—always near the geographical center of the United States. 1959 saw the Nationals move to Detroit, with the express intent of exposing this growing sport to the eyes of the automotive industries, right in their own front yard. Huge success of the initial BIG GO at Detroit was enough to prompt a second year's presentation there, even though the location was a little too far east for western contestants.

Original Nat'l Championship Drag Races, presented by NHRA, 1955 at Great Bend, Kan. Turnout far exceeded expectations.

Kansas City, Mo., site of 1956 Nationals, held at a new strip which the Safari crew had helped to construct months earlier.

1960's Nationals at Detroit was even more of a success from all standpoints, and the event gained even more prestige in the eyes of the automotive-interests world.

To balance the rather eastern location of the Nationals, NHRA decided to produce a second major event, this time "out west." The natural solution was to move its Winternationals, presented in Florida in February, 1960, to a suitable western location in February, 1961. The spot chosen was the Los Angeles County Fair Grounds, site of Pomona's well known drag strip. The '61 Winternationals far exceeded anyone's wildest expectations—so many people turned out that the gates had to be closed to spectators at 1:30 p.m., and traffic was choked on roads and highways all around the Fair Grounds. Competition was tremendous, with contestants on hand from most of the western states area and quite a few from eastern locales. NHRA thus had its Nationals and its Winternationals—one eastern and the other western—each spaced six months from the other and both affording the ultimate in drag racing interest and color.

The 1961 Nationals were moved again, this time to Indianapolis, Indiana, at the "Capitol of Auto Racing." A brand new drag strip had been constructed as a part of a vast new racing enterprise known as Indianapolis Raceway Park. Located just six miles northwest of the famed 500 Speedway, the I.R.P. facility included the most up-to-date accommodations for drag racing, plus a sports car road racing course and an asphalt paved oval racing circuit. NHRA considered this the ultimate answer to its increasing need for a site befitting the Nationals, and I.R.P. welcomed the BIG GO to its events agenda.

But the "rather eastern location" of Indianapolis failed to dampen the enthusiasm of contestants in other parts of the country—a total of 40 states were represented among the approximately 800 cars entered in the '61 Nationals. More than 250 of these were in the Gas coupe/sedan classes, almost 200 were Stock class entries, and over 100 were all-out Dragsters. Each of the total 62 classes was well equipped for active competition as the entries reached their closing date and the sport's biggest field of hot rodders headed for the site of their BIG, BIG GO.

Other phases of activity, too, had been developing in NHRA circles, aimed at adding more significance to the sport in general. Among these was a new, authentic National Records system in which true performance accomplishment was rewarded with honest recognition. Each of the NHRA Divisions was authorized an official Record Certification Crew, which was entrusted with the responsibility for making certain the official records established in its respective Division were above reproach. This required preliminary planning, training and special equipment for testing fuels, engine legalities, wind velocity, strip gradients, etc., all vital to the determination of record authenticity.

Above Left—Trailblazing members of first Safari crew were, lef to right, Bud Coons, Bud Evans, Rick Rickman, Chick Cannon. Above—New Safari crew, still headed by Coons. Safety Safari produced a total of 36 major championship events in 3 seasons. Left—1961 Plymouth station wagons, assigned to NHRA's field divisions, were used to replace the single-team Safari.

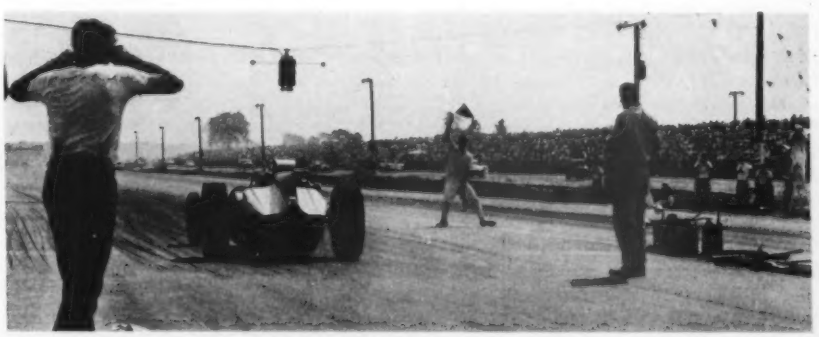
Typical of the way NHRA representatives helped assist in development of new drag strips is this scene at Amarillo, Texas.

Oklahoma City, location of 1957 and 1958 Nationals. Overflow crowd thronged State Fairgrounds where strip was located.

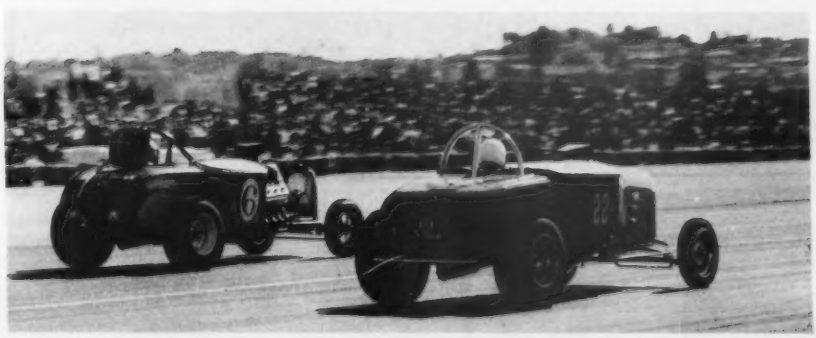
Double-check timing systems were required, to ensure accuracy of elapsed times and top speed records, plus which procedures were established to provide verification that the required conditions were complied with throughout. Once and for all, the drag racing sport had finally achieved an established system through which its performances coast-to-coast could be measured and compared.

To add further interest to each season's competition, NHRA produced its own version of official points system, in which the average contestant could compete for high honors based on his consistency in winning first place in his class at sanctioned events in and around his own area. The points season was limited to the months in which drag strips everywhere are generally in operation, and the sport joined others in the sporting world by declaring its top points winner the World Champion. By distinguishing its runners-up as the "Top Ten," drag racing's points program became one of the most popular phases of the competition year. Major awards, such as new automobiles, were posted for the Points Champs in both the hot competition division and the stock cars division, with the Top Ten designation divided equally among the stock and hot competition points winners. Basing points on class wins at regular weekend events, class wins and eliminator wins at special bonus points events and the big bonus points grand finale at the Nationals, the World Champion points program has grown in stature until it now rates as the sport's ultimate honor. Of special value is the fact that one class had equally as good a chance as any of the others in having its top winner the World Champ.

Although drag racing has encompassed much of the time and energies of National Hot Rod Association, there is a second side of NHRA that is of even greater importance, even though it has been overshadowed in the amount of acclamation it has received in recent years. This, of course, is the "keystone" element—the car clubs and their everyday activities. With drag racing well organized and making excellent progress, NHRA has established a separate section known as the Car Club Division whose sole responsibility is to assist with the development of a successful future for car clubs and their activities. The official Charter Club program instituted by NHRA has done much to stimulate this field, and the recent addition of the new Car Club Division has launched a whole new era of interest in hot rodding. This brings us back to paragraph one, and the Association's primary objective, which completes the historical cycle and adds the assurance that hot rodding is as NHRA does. It's a big responsibility for any organization to carry, but the continuing accomplishment can be summed up in the terms of two slogans of the National Hot Rod Association—"Dedicated to Safety" and "Ingenuity In Action." ■

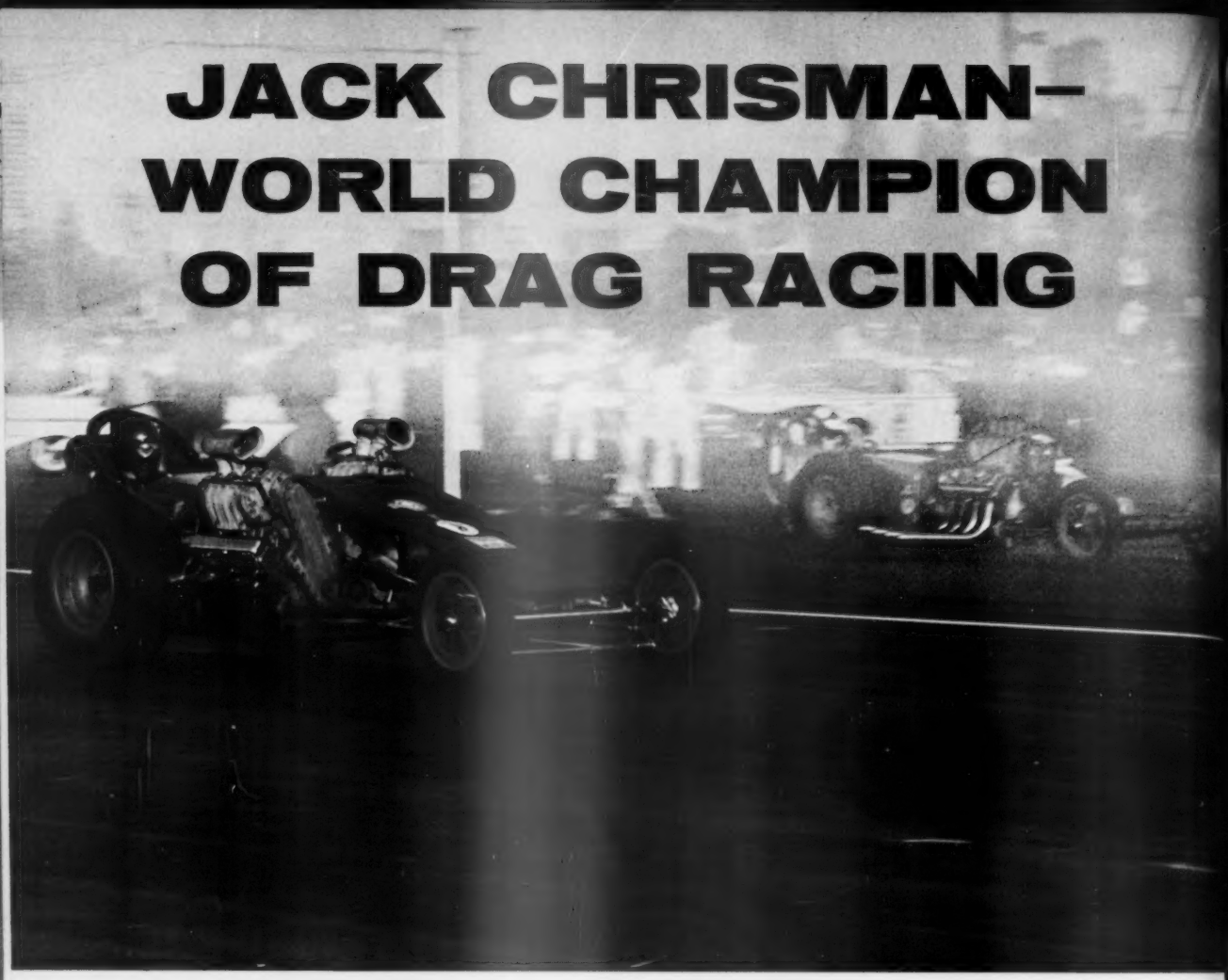


1959, 1960 saw Nationals move to Detroit. The Motor City was given its first look at drag racing on a major scale. The event, extremely successful both years.



1961 Winternationals, presented by NHRA, Los Angeles County Fairgrounds, Pomona, Calif., drew fabulous crowds.

JACK CHRISMAN—WORLD CHAMPION OF DRAG RACING



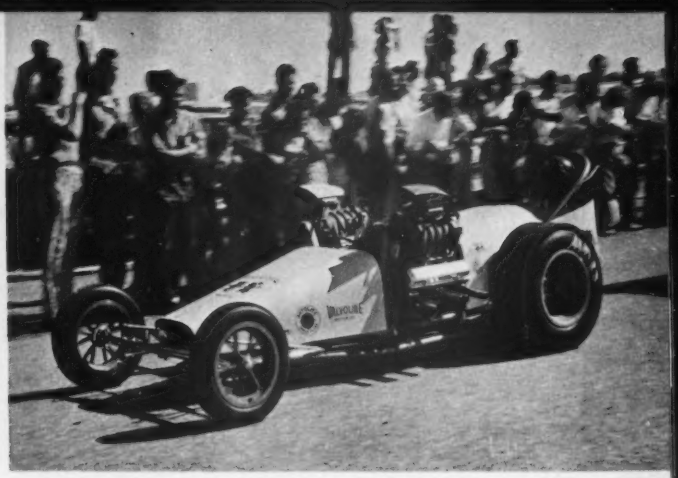
THE RED 1940 Ford coupe makes a hard right turn into a small root beer drive-in and the driver jumps on the brakes, switches off the ignition and is out of the door all in one jerking motion. Two long steps and he is yelling into the nearby chopped custom, "Hey, he's here! I just saw him pull into the motel over on Main Street. Man, oh man, you ought to see that rig. The biggest slicks in the world. And those engines, superchargers up to here, boy . . ." A typical reaction in a typical town near a typical drag strip. Jack Chrisman was in town. Jack Chrisman, the 32-year-old father of two from Southern California who was touring the United States in an all-out bid for the NHRA World Points Drag Racing Championship.

Chrisman was still fresh from his resounding triumph at the '61 Winternationals when the thought of winning the points crown began to take shape. During the early part of '61 he had sold his service station. For some time Jack had been driving the twin Chevrolet engined dragster running out of Howard Johanssen's speed shop, and subsequent discussion between Howard and Chrisman convinced them both that a good tour by a really representative drag team was certainly in the offing. Of course, finances were to be considered, and from the start this trek across thousands and thousands of miles was not to be a subsidized affair. In essence, Jack was to be on his own.

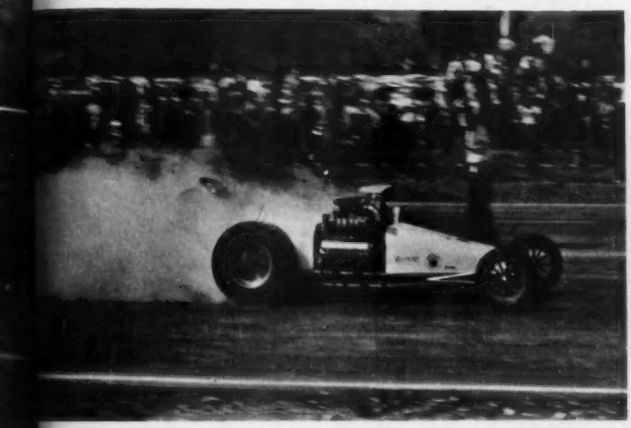
Once the decision to make a concentrated effort to win the World Champ title was cast, Jack set about in earnest to prepare the Howard twin-Chevy Special. Realizing that a very small crew would be necessary to help keep expenses down, he went one step further and selected just one crewman . . . Jerry Johanssen. Jerry is Howard's son and has been around hot racing equipment since he was knee high to a belly tank. He was the ideal choice for the trip. In the Spring of '61, Jack and Jerry departed Los Angeles on a trip that was to take them across the country many times. They were to run at dozens of different drag strips, meet thousands of people, become the prime target for every fast dragster pilot in the nation. They would meet with and best all kinds of conditions, from poorly constructed drag strips to bad weather. But all the time they were making new friends and proving to the world the inherent qualities of top notch hot rodders.

And the fame of Jack Chrisman spread like wildfire. Oh, he wasn't exactly unknown. Not by a long shot. He had been one of the pioneer drivers of drag racing cars. His experience included nearly every form of drag machine, from rear engined bears to Model A sedans to the new twin-engined monsters. Among the drag fraternity he was considered as one of the most capable of high horsepower drag pilots. But the mere fact that this fellow—who had

Twin engined Howard Cam Special gave Chrisman title of Top Eliminator in Winternationals at Pomona, Cal., Fairgrounds.



Car used Howard roller cams tuned by 19-year-old mechanic J. Johanssen. Chrisman drove 24,000 miles on record circuit.



When Chrisman floorboards it, things happen fast! Family man Jack is one of best liked drivers in business of record setting.

George Hurst (center) presented new '61 hardtop T-Bird prize to Chrisman at 1961 Nationals. Mechanic J. Johanssen, right.

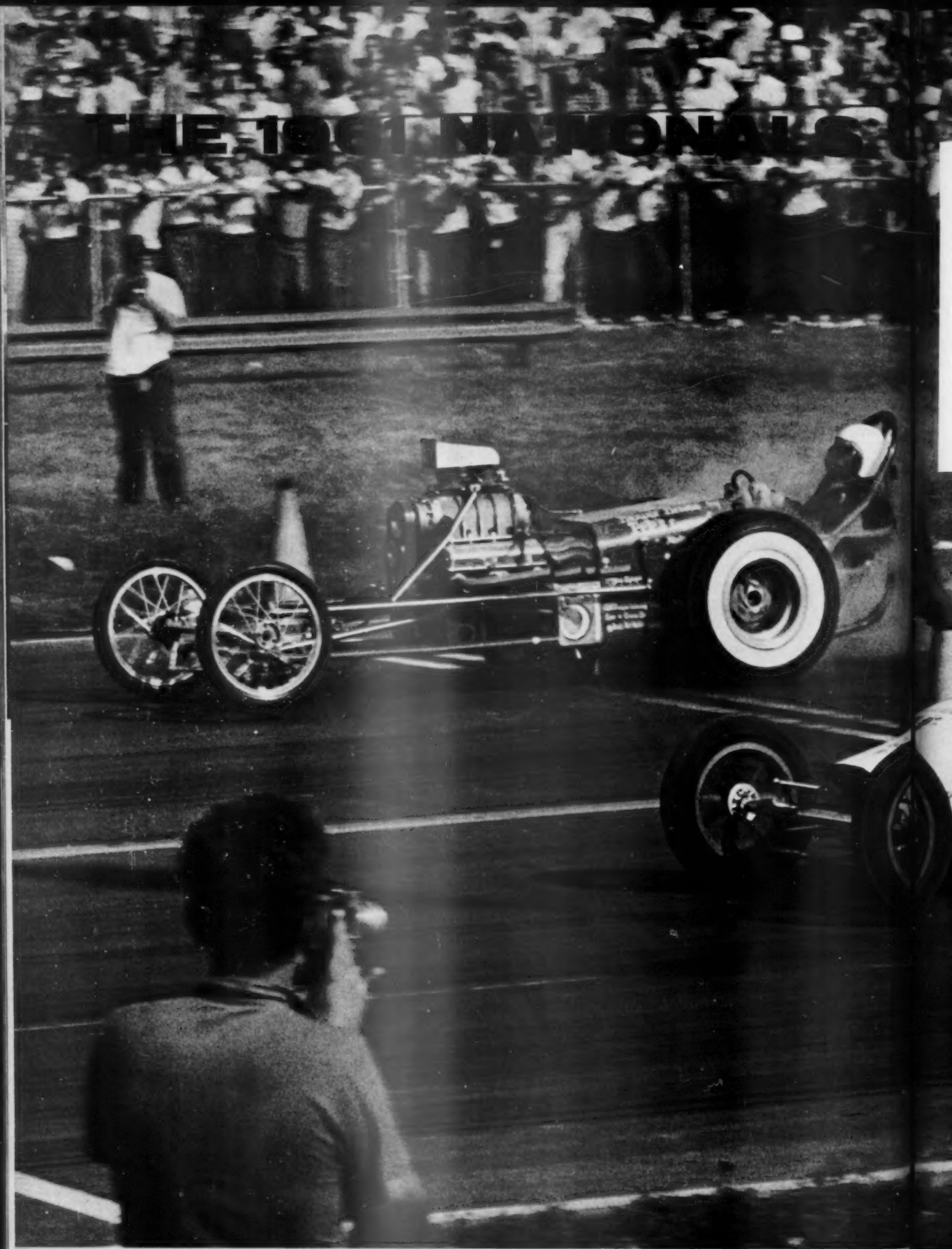
heretofore been only a name printed in a book or newspaper—was finally to be seen in action spread enthusiasm everywhere. Wherever he went, so did the spectators. And true to his style, he talked to everyone. The hot dog, the novice, the average hot rodder.

As so many of the really good dragster drivers, Chrisman is quick to point out to fledgling quarter-mile enthusiasts that there is absolutely no substitute for experience. You just don't step out of the family sedan and into a screaming rail capable of over 160 mph. It takes hundreds of hours to train a good Air Force pilot, and training is certainly necessary for handlers of hot equipment. If they want to be winners, that is.

Well, the trip is over. Jack Chrisman has accomplished what he set out to do. After being named to the exclusive ranks of Motor Life Magazine's "Racing Men Of The Year" panel (HRM, June 1961), winning the World Points Championship might seem anti-climactic. But not in the least. Jack won the drag crown with 550 points, all tallied over the best drag racing competition in the world. But the winning of the title has brought on new problems. What next? Where to go from here? That question hasn't been answered as yet, but one thing you can be certain of. When the time for trophy presentation comes around, a sinewy black haired fellow will be standing in line front-row-center. Jack Chrisman—The World Champion of Drag Racing 1961!

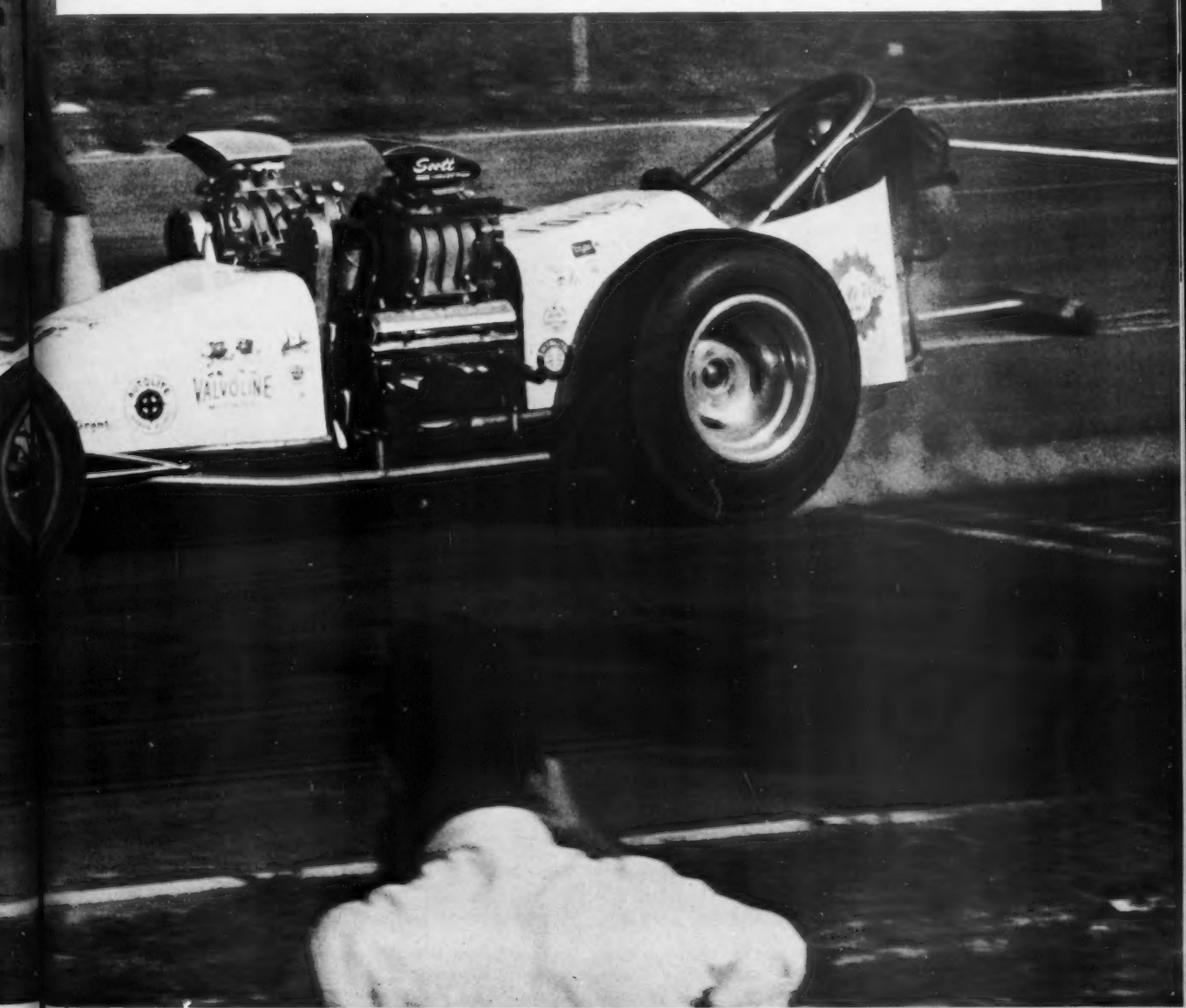
TEX SMITH

Jack Chrisman, with his Howard Cam Special, looking it over before making bid for record run. He has years of experience.


THE 19 NATIONALS

FROM the cotton fields of central Georgia to drag racing's alpine heights is a long, long way. Especially so if measured in terms of experience and knowledge necessary to gain such exclusive rank. For Pete Robinson of Atlanta, Georgia, that distance is one-quarter of a mile; 1320 precisely measured feet; approximately three city blocks; just a bit more than four times the length of a football field. Small as this measurement might seem to a space-age mind, it is all that was required to catapult a virtually unknown automotive enthusiast to the very pinnacle of hot rodding success. The thin ribbon of macadam called a drag strip hardly seems the proper place for a Cinderella story, but the accounting of Mr. Pete Robinson and the 1961 edition of the National Championship Drags is just such a tale. But to recall the story in its most accurate form, let's start at the beginning.

Officials of the National Hot Rod Association had announced early in 1961 that the 7th annual championship drags would be held on the newly completed Indianapolis Raceway Park near the famous '500' Speedway. Everyone connected with the sport was enthusiastic over the news since the newly constructed IRP facility was recognized as one of the finest racing plants ever built in the U.S. IRP officials had built the raceway to encompass all aspects of motor racing, but they had cast a discerning eye



at contemporary drag strips and decided that theirs was to be a truly professional facility.

Drag enthusiasts must have approved highly of the mid-Indiana location too, as they began early to bombard the NHRA headquarters with applications to race and requests for information. It was soon apparent that the '61 Nationals was drawing far and away more interest than any previous meet. A great deal of emphasis had been placed on competition at the Nationals by contestants in the points competition system. Jack Chrisman had outdistanced all challengers to the crown in the Competition Car section and was assured of winning the beautiful white Ford Thunderbird to be given the points king by the Hurst-Campbell people. Bruce Morgan knew he had a good chance to clinch the Stock title and a 1961 Pontiac from Hurst, which had been set up for drag racing by Royal Pontiac of Michigan. But still, the rest of the competitors wanted to win as many points as possible and perhaps ease up higher in the standings. Since the 1959 and 1960 versions of the annual affair had been held in Detroit and had served prime notice to the automotive industry that drag racing and hot rodding were more than the tinkering of irresponsible youth, it was known that the industry was more than casually interested in how the stock class cars stacked up. A lot had been done in various parts of the nation to make production automobiles perform in superb fashion and Indy would be the showdown.

By the time the August entry deadline date rolled around more than forty states were represented in the Big Go and over 900 top competition cars were being prepared to do battle. Of this total, always strong Ohio led the field with nearly 300 entries, but California was represented with the largest contingent of Golden State vehicles ever to sign up at a Nationals. Some mighty big racing was brewing.

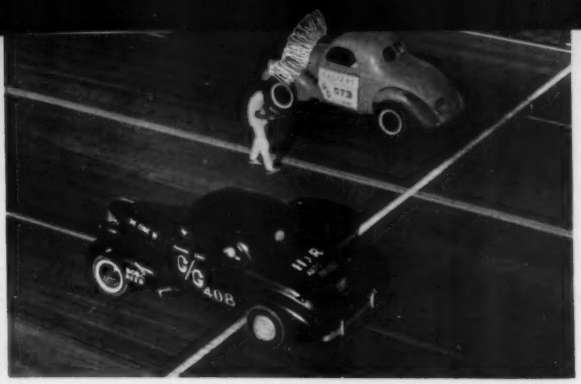
The Midwest is famous for its lush green landscape and fertile soil, a reputation gained through the helpful cooperation of the rainmakers. The few days before the opening of the Nationals on Friday, September 1st, had been dry and clear, but nearly everyone was keeping one hand on an umbrella. Indiana weather can be rather unpredictable. A warm front did slide lazily into the Hoosier state over the long weekend, but Indianapolis escaped any noticeable moisture. All moisture, that is, except that contained in the high humidity readings.

Activity at the IRP strip got underway on Thursday with the opening of the technical inspection area under the direction of Tech Director Farmer Dismucke and his expert crew of tech inspectors assembled from across the country. No actual Nationals racing was planned for that day but the Indianapolis Dodge Dealers Association had arranged to present their highly successful Dodge Performance Day for press representatives from the Midwest. Chrondek clocks in the air-conditioned three-story D-A Lubricants timing tower were worked overtime as members of the working press tried to run the wheels off the various Dodge and Lancer cars assembled for the purpose. These vehicles were all race prepared to one degree or another and the newspapermen and broadcasters were treated to a first-hand feel of organized dragging. A feel they liked extremely well, we might add.

As always, the tech inspection lines had a good share of cars rejected. At a really big drag meet there are bound to be those people who will prepare a car with the thought of just getting by, or who think they can talk the inspectors into letting the car pass. Some fellows always end up getting their feelings hurt. Out of an estimated 18,000 runs at this year's Nationals, only three accident possibilities were recorded—two times when cars wiped out the center timing lights at the finish line and one when a drag chute failed to open properly and the car had to use the extended stopping area. We have noticed that whenever the experienced race driver is advised of a potential danger area in his car he fixes it. It always seems to be the would-be hot dog who protests most before fixing his car.

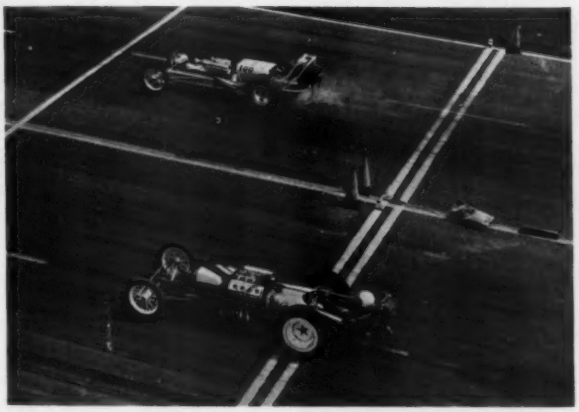
Friday morning found the pits and staging area filled to overflowing with cars and drivers anxious to get a run at the new raceway asphalt. The strip was acknowledged to be in excellent shape. Announcers Bernie Partridge and Stan Adams of California, and Dave McClelland of Shreveport, Louisiana, handled the chores of keeping the vast crowd informed of all details over the truly outstanding public address system that is part and parcel of the IRP facilities. NHRA uses an interesting event control operation that has proven very effective at their big meets. A vast communication network connects all portions of the strip to one central office called Monitor. Dale Ham of Amarillo, Texas, usually handles Monitor and passes on to the announcer/timing booth only that extra-curricular information deemed absolutely necessary to be broadcast. Otherwise, all action is instigated and settled in Monitor. This innovation is not entirely new in drag racing but never have we seen it developed to such a degree. In fact, the sonic-paced action of this year's event prompted one of the best Midwest sports writers to comment in his newspaper that perhaps some of the major professional racing organizations should take a few lessons on race presentation from NHRA.

The initial day of drag racing at all past Nationals has usually been considered more of a convenience to the participant than to spectators, but a crowd of from 10-12 thousand people was lined up along the course for the Friday get-acquainted runs. By two o'clock in the afternoon everyone present was sure that the winning elapsed time and top speed for the 1961 Top Eliminator title would be somewhere in the 8 second bracket and high 170's. Dick Wells, Editor of National Dragster, was overheard discussing the coming events with Jim Nelson of the Dragmaster Company before the strip had opened up on Friday morning. Jim was quite specific when he told Wells that before a car had even run on the excellent asphalt surface one thing was certain—the winning car would most definitely have an elapsed time lower than 8.95 seconds. Such predictions are not commonplace at a championship event, primarily because competitors just usually don't have the necessary time to get their machinery to top performance while away from home. It is usually agreed that a championship meet will be won with elapsed times and top speeds just a shade poorer than what is currently the trend across the country. But the '61 Big Go was going to be different. Already several favorites had cropped up. One of them being the big healthy AA/D from the stable of Gene Adams and Tom McKewen. This team was fresh from some fantastic upper 8-second wins on the West Coast and they captured best performance nod on Friday with a top speed of 170.45 mph in 9.03 seconds. You will recall that Gene Adams was a part of last year's winning Albertson Olds combination and his new car gave every indication that power was no problem whatsoever. Right behind the Adams–McKewen car in the top performance spotlight came Jack Chrisman with an e.t. of 9.22 at 165.44 mph, Mickey Thompson with an e.t. of 9.57 and top speed of 156.52, Dode Martin's Two Thing at 169.17 mph in 9.10 seconds, and a whole rash of going double and single A class dragsters. Big gun in the B Dragster class was fast cropping up in the form of Marino Monjure of Louisiana

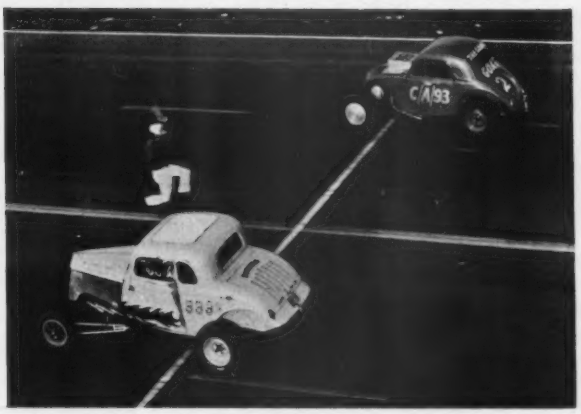


Pete McNichols '61 Valiant powered Willys was a favorite, bested Bob Reed's Chevy six for popular win in G/Gas class.

Top Eliminator Pete Robinson pulls his single engine AA/D off against Eddie Hill's twin-Pontiac rail. Robinson won.



Dean Moon's A/Dragster, far side, ran Dean Corey from Fostoria, Ohio, in elims, ended up winning the class trophy.



The blue '34 coupe running out of Al's Speed Shop in Ill., lost to Billie Rasmussan's Fiat-Chevy in a tight finish.

Jim Bell brought his unusual C/Altered Metropolitan up from Florida to run but had some small teething problems.

Bud Faubel uncorked his '61 Fury in an attempt to stem Ray Christian's Plymouth but to no avail. Ray won A/SA class.

The high standing Bach's Body Shop A/Gas blown Willys is a good example of attempt to get best weight transfer.

Winner in the hotly contested C/Gas Supercharged class was the San Fernando, California, Willys of Pittman-Edwards.

with speeds in the neighborhood of 136 mph and e.t.'s in the mid-10's. Wally Markert from Massachusetts proclaimed early that he intended to maintain his domination of C Dragster class with runs of over 130 mph in the very low 11 second category. Al Carter of Oklahoma City had been pushing his throbbing Chrysler 'T' through the lights in the 150 mph range and Walt Knoch's new Puffer A/Altered was flirting with the records on runs of 154-plus mph in the high 10's.

But even with all these fine times in the hot car categories, who do you think was getting all the applause from the crowd? You guessed it, the stocks. And little wonder, because Hayden Proffitt had been pushing Mickey Thompson's Optional/Super Stock Pontiac through the lights with some good recordings and on one occasion got a top speed of 112.21 mph in 13.07 seconds. Right behind Hayden came his West Coast nemesis Don Nicholson in a 409 Chevy with times of 110.29 mph in 13.25 seconds. Bunched right behind Proffitt and Nicholson were at least twenty top running Pontiacs, Fords, Chevys and Dodges, each one capable of pulling off a big upset stock win.

While activity on the strip itself was fast paced all day long, repairs and technical corrections in the pits were even more apparent. The Hurst-Campbell Corporation (manufacturers of dual pattern floor shifts) had set up a big van machine shop in the pit area and you could hardly get near the place for the rodders fixing things. The local Indianapolis distributor for the Lincoln Electric Company brought out some portable arc welding equipment to be used out of the van for necessary welding repairs and fabrication of such things as roll bars and scattershields. As in the past, the major manufacturing firms directly connected with automotive racing were on hand to lend assistance to the competitors. Spark plug problems were handled by Dick Jones and Skip Mason of Champion and Fran Hernandez, Director of Racing for Electric Autolite Company. Both of these companies had a top notch crew of experts on hand to diagnose plug problems and offer solutions. Jack Martin of D-A Lubricants was present to help everyone with oil problems and was constantly filling requests for D-A Speed Sport oil.

Friday night there were just two things to do; go to the Custom Car Show or hide. Being tossed into the motel swimming pool, clothes and all, was the order of the day.

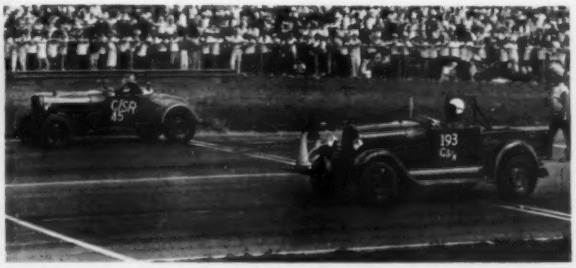
The strip opened early Saturday morning and contestants who didn't get to the park early found themselves ensnarled in a huge traffic jam that stretched all the way from the vicinity of the '500' track several miles to the little town of Clermont where the IRP track is located. Quite apparently no one but the NHRA officials was prepared for the huge crowd that descended upon the race course. No major problem arose however, and the stands that line the quarter-mile were soon filled to capacity. Saturday was scheduled for stock class eliminations and normal time trial runs for all hot cars. As soon as the stock eliminations began it was certain that this Indianapolis crowd was overly conscious of the performance standards of drag racing stockers. Even in the lower stock classes which normally go unnoticed the crowd reacted to each favorite with wild enthusiasm. One example was in the hotly contested H/Stock class where a 1960 Valiant cropped up as a potential winner. After just one elimination run the little white compact had suddenly become the favorite and it lived up to all expectations, pulling owner Charles Gross of St. Claire Shores, Michigan, to a win in 15.96 seconds at 85.06 mph. Another compact in the winner's circle was the '61 Corvair of Dave Bassett in J/Stock class. Its winning speed was 78.46 mph in 17.44 seconds. Just goes to prove the new little cars can hold their own against their larger counterparts of past years if they're tuned and driven right.

One indication of how a cheering crowd can vacillate back and forth between favorites in dragging is exemplified by the results of the hot B/Stock class. Bruce Morgan of

The big race for all the marbles in B/Stock class pitted Richard Hilt, left, and Bruce Morgan. Hilt won the tight contest.

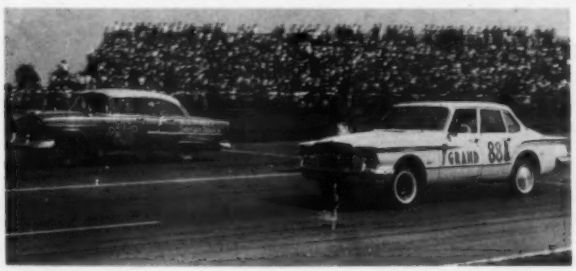
A close race but Willis Ragsdale of Texas beat Milton Potter's Tennessee A/Street Roadster for Little Eliminator.

Two well known cars, the Ratican-Jackson-Stearns-Reath Fiat and Charles Scott's Bantam. The R-J-S-R A/Altered won.



Lim Lightcap's GMC powered roadster, left, beat Jim Shepherd's GMC 'A' pickup for the title in C/Street Roadster.

In Top Eliminator runs Bob Carroll's A/Competition coupe out-pulled the Starkey-Jent Special B/Dragster from Ohio.



The Bill Bradley Chevy found things a bit hot in H/Stock class, lost to Charles Gross in a 7000 rpm white Valiant.

The new A/Altered Fiat coupe belonging to Walt Knoch turned in some amazing performances in 155-plus mph range.

San Gabriel, California, had arrived at the raceway as a favorite to take the class and possibly one of the Stock Eliminator titles. Throughout the running of the B/Stock class the crowd was with first one and then another favorite, until the final two cars to have a bit of a go at it were Morgan and Richard Hilt of Lafayette, Indiana. The final run was a close one all the way and had the entire spectator turnout, as well as the pit area people, straining to see who would win. Morgan had a possible World Stock Points Championship riding on the run, but it was Hilt to the lights first and the roar that greeted the victory was overwhelming. The little fellow had won, much to the delight of the crowd. As it turned out, Morgan had enough points accumulated prior to the race to win the championship and its consequent '61 Pontiac prize posted by the Hurst-Campbell Company. However, during the technical tear-down of the cars to again certify their legality, Hilt's was discovered to have improper valve spring retainers and was technically disqualified. This might seem a minor point to some, but stock means stock, and any deviation whatsoever should not be allowed. Even with the disqualification this boy Hilt bears watching as a big threat in next year's B/Stock class.

It is nearly impossible to put down in printed words the effervescent and omnipresent thrill that sweeps through a drag racing crowd when the big stock cars come to the line. With exhaust stacks open the cars sound a great deal like the popular Gas class machines, and their performance is nothing short of amazing. Unless you are lucky enough to live in a section of the country where the big super-sized stocks are prevalent, then you can't really understand what kind of popularity they breed. Primarily, the big stocks give amazing performance but they still provide those looking on with some sort of personal identification. If a big Ford wins, the spectators with Fords feel a bit of camaraderie with the winner. The same with the Chevys, the Pontiacs, etc. The variety of cars entered in the top stock classes was really unusual this year. There were the normal number of Chevrolets and Pontiacs but more Fords than ever were lurking in the staging lanes with just enough Dodges and Plymouths to make things very interesting.

As it turned out, Hayden Proffitt took the big Optional/Super Stock class win with a top speed of 110.29 in 12.55 seconds. On this winning run there was a good steady 7-9 mph quartering tail wind, but anyone who races a lot will tell you that a tail wind doesn't do a great deal of good on short sprints. However you look at it, though, when a big stocker gets down in the 12 second bracket, brother, it's hauling. Proffitt hadn't even turned and headed back toward the pits on this 12 second run when NHRA's fuel expert Dean Hill was under the hood taking a gas sample with a hypodermic needle directly from the carb dump tubes. Dean told us later that the gas checked out perfectly. In fact, he said that he only found a trace of exotic fuel in one car and that was because the car was apparently run on a mixture containing Hydrazine some time before the Nationals. Super Stock class was taken by Arnold Beswick of Mor-

rison, Illinois, and Don Turner of Fortville, Indiana, won the Optional/Stock Automatic awards. Lloyd and Carol Cox of Whittier, California, picked up the glory in Super Stock Automatic class.

A quick check in the final results listed elsewhere in this story will show that Chevrolets predominated the stock classes with Pontiacs close behind, but there were some deviations with Ford and Dodge coming home the winner. We feel safe in predicting that a greater number of drag trophies will go to these last two marques during the 1962 season. Especially so with Ford's new four-speed transmission and rumored goodies for the Dodge.

The fifty top elapsed times in the stock car classes were scheduled for a gala affair on Monday, with the winner of the contest to be crowned Stock Eliminator. Now, this is no insignificant title. In fact, the automotive industry looks toward this particular honor with more than a passing interest.

With the stock eliminations out of the way, everyone could return their attention to the competition cars. Before anyone fully realized it, four big dragsters had tied for the top speed of the meet honor at 170.45 mph. They were Joe Shubeck of Lakewood, Ohio, Dode Martin of California, the California car of McKewen-Adams, and Texan Eddie Hill's twin-engined Pontiac. All these cars were potential winners but Jack Chrisman was right behind with an e.t. of 9.01 at 169-plus mph. All during the day the elapsed times had been skirting on 9 seconds flat. We had personally been estimating top times of around 179-180 and e.t.'s of about 8.95 as the best of the meet. So far we felt fairly secure in our estimates. But then we got wind from Event Director Ed Eaton about an amazing single engine car that had been posting times in the middle 8 second bracket. It is the policy of the NHRA crew never to announce times that are questionable until every little bit of doubt has been removed, consequently these times had remained unannounced all day Saturday.

Sunday dawned nice and clear and so apparently did the spectator crowd. The Raceway gates were opened earlier, at 6 a.m., and by the time for the runs to get underway at 8 a.m., nearly everyone was seated. Some 40,000 everyones. Right off the bat a little red Dragmaster chassis single engine car set the meet in a turmoil. The loudspeaker blandly crackled out that the car had just turned an elapsed time of 8.52 seconds. Pandemonium in the pits! Previously sparsely occupied dragster staging lanes suddenly bristled with equipment itching to get a run. Perhaps the rail boys thought that some atmospheric condition was helping the single engine car, or maybe they just wanted to take a crack at the 9 second barrier. Whatever the reason, Pete Robinson of Atlanta, Georgia had served notice in a big way that he had every intention of capturing the goodies at the '61 National Championships.

Eliminations in the hot car classes were something to behold, what with the tremendous turnout of cars. The only class that went virtually uncontested was A/Modified Sports, which went to the Mayor of the City of Industry, Calif., Mr. Sam Parriott. Of the nine hundred plus entries,

Don Nicholson's Chevy, 409 variety, had a tough time beating Arnold Beswick's torrid Pontiac, was later disqualified.

over one hundred were dragsters. The Gas classes accounted for nearly three hundred entries alone. Incidentally, the big story in these classes was the extreme rpm being cranked up by all kinds of engines. The Chevrolets were turning on, but so were the Pontiacs, Fords, Studes, Valiants, etc. At the same time that engines were twisting past the 7500 rpm mark, there were relatively few clutch/pressure plate failures. This may be attributed directly to the advancements made by flywheel and clutch equipment manufacturers in the past year.

The first round in dragster eliminations went about as expected with all the top runners emerging on top. The real story was the fact that all the big cars had cranked winning elapsed times well into the 8 second range. Working through the classes by the round-robin system, some of the favorites in the other classes were soon dropped by the wayside. One such victim was the wicked A/Altered of Walt Knoch. Consistently cranking e.t.'s in the 10's with top speeds of near 160 mph, Knoch's Puffer was tabbed as the car to beat. The team of Ratican-Jackson-Stearns-Reath of Hollwood, Calif. accepted the challenge and proceeded to turn back the big roaring Fiat to win the class. As a sidelight, it seems that you're a nobody anymore in the Altered class unless you run a Fiat body. Actually, they are rather slippery and lend themselves readily to the class.

But Walt Knoch wasn't shut completely out of the picture because his latest car, a torrid A/Roadster, hauled home the bacon in that class by storming to the victory lane in 10.20 seconds at 155.97 mph. Sure would make a crazy street machine for chugging through drive-in restaurants.

An interesting battle developed in the D/Dragster class when some really hard-charging cars turned out to joust. When the smoke had cleared from rubber churned up by the flatheads and inline sixes and eights, Stan Lomelino of Veiden, Illinois, reigned supreme with his '46 Ford engined rail. Even so, George Herbert of London, Ontario, Canada, picked up a brand new slant-six Lancer engine given by the Indianapolis Dodge Dealers Ass'n. George won the prize for twisting his GMC to the lowest elapsed time of the meet for an unblown six cylinder car.

Back in the Altered classes, Waldo Hirshfield and K.S. Pittman of Calif., were wading through a sea of competition to pick up B/Altered honors in their B and M Hydro equipped Fiat. An amazing story was written in the C/Altered listings when Billie Rasmussan of Texas won his fifth straight Nationals class trophy. And that's not all. Billie usually helps run the classification scales at the Big Go and doesn't get any chance at tune-up runs. This year he was back with a Fiat body surrounding his '53 Dodge engine. Since he was busy trying to help get everyone else on the strip, a couple of acquaintances agreed to run the car.

While all the action was taking place out on the strip, an interesting aside was shaping up behind the pit area. The IRP facility encompasses a paved road racing course and the raceway officials had requested that no competitors try their hand at sporty car type handling. The raceway group and Ed Eaton proved that they meant what they said when they ejected one of the better known stock car racing teams from the event for spinning doughnuts in the sport car track area.

Back at the ranch, eliminations had proceeded smoothly until the big question was whether or not the little fellow from the South could stave off all the big guns trained on him. Apparently he was going to try, because as the chips happened to fall, he drew a rugged opponent every time out. The elapsed times continued to stay in the 8 second area. Although Robinson was looking more and more like the Cinderella driver, the other boys were doing great too. A notable example was the driving skill of Tom McKewen. On one particular run Tom happened to draw a very fast

The Adams-McKewen car had to do some real hauling to beat out Joe Shubeck, far side, in the AA/Dragster eliminations.

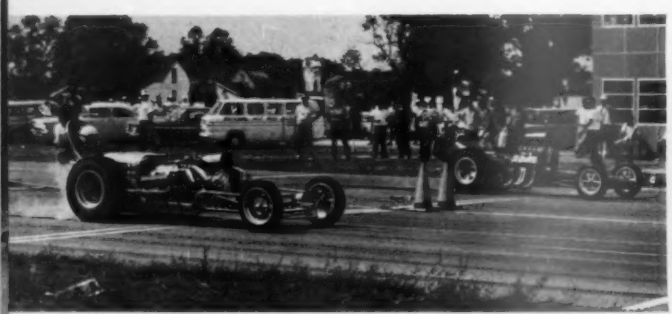
competitor. Off the line Tom's car broke loose severely but through masterful driving technique he managed to push the super-powerful Olds rig to a win. That particular incident placed him squarely in contention with Robinson as the favorite with the screaming crowd. But Sneaky Pete was hanging on. On one of the runs, Pete happened to draw Eddie Hill. Hill is no newcomer to low e.t.'s and meant to give Robinson a real run for the money. Hill came out first and paced Pete to the lights until the terrifying little 352 cubic inch Crane cammed Chevrolet uncorked. Hill's car was equipped with four slicks on the rear and had actually *melted* the asphalt at the starting line on the take off.

When Jack Chrisman came up to the line against Robinson you just knew that here was to be a race. Both cars left the line hard, the big twin-Chevy Howard Cam Special lifting the front wheels up off the ground for some distance. Chrisman was way out in front for the first part of the drag, but near the lights the Georgia fast freight went shucking home. We asked Jack later how come he got beat. His answer, "That little car just plain outran me." Brother, you have to have some horsepower to outrun the twin-engined cars on the top end. The final run for A/Dragster had come up in the meantime and Dean Moon's "Mooneyes" dragster blasted home the winner with speeds near 160 mph and with e.t.'s in the mid 9's.

Finally the big run was at hand. The two remaining cars in AA/D class were Robinson and the McKewen-Adams car. At the flick of the starter's flag McKewen was out of the hole with the big slicks throwing rubber everywhere and the front wheels waving to the crowd. As usual, Robinson was last out of the hole, but he wasn't losing any ground on the big, blown Olds. The entire area was a sea of waving hats and cheering voices. Previously it had been learned that the Adams-McKewen car had broken a cylinder wall and was running without the benefit of any water at all. You wouldn't have known it the way McKewen had his foot stuck in the injectors, but the same old story proved true. It was the Georgia peach at the lights, winning with a top speed of 170.77 mph in 8.86 seconds. Just about that time the value of the Confederate dollar went up considerably. The South was on the march again. When the strip was officially closed Sunday evening, Robinson had set the lowest e.t. and Dode Martin had claimed a top speed with the Dragmaster Two Thing at 177.16 mph. Sunday night found members of the HRM staff and the 1960-1961 members of the Top Ten Club enjoying conversation and prime rib at their first annual club banquet.

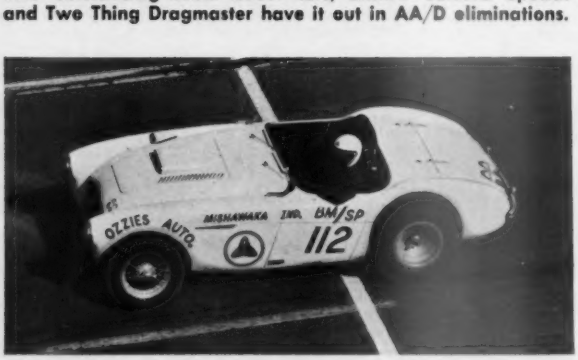
At various times throughout the meet, Tommy Ivo fired up his big four-engined Valvoline special for exhibition runs. Tommy and his Buick-engined rail were present at the festivities through the courtesy of Buick and Valvoline and appeared nightly at the National Champion Custom Car Show. He was a double threat at the show; the girls wanted his autograph as a TV star and the boys wanted to know about Buick engines and Valvoline oil.

Right now might be a good time to pass along a little information about this amazing whiz kid from central Georgia. Pete Robinson had competed at three previous Nationals with a '40 Ford coupe. Along during the spring of '61, he and and friend Bill Word decided to build a dragster to advertise Pete's new competition supercharger drive and quick-change for Chevrolet differentials. First thing they did was buy a Dragmaster chassis and settle on Chevrolet for power. The two fellows told us that they really didn't have any overwhelming secrets in making their car run. They said with disarming frankness that all they'd done is put lots of compression ratio in the engine and found the right gear ratio for the rear end. In fact, the high



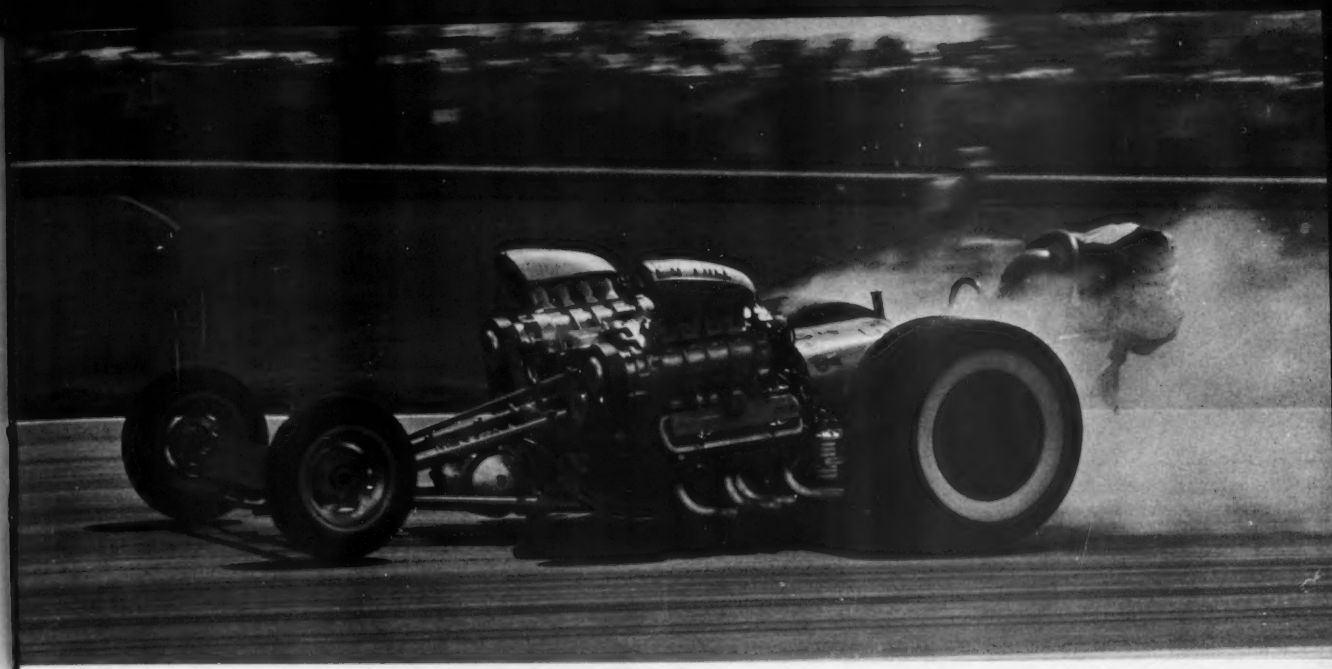
Two outstanding West Coast cars, Ernie's Camera Special and Two Thing Dragmaster have it out in AA/D eliminations.

An overall view of the staging lanes and pit area of the Nationals. A/MR winner Dave Skeans is firing up for a run.



Eventual winner in the B/Modified Sports section was Jerry Osborn of South Bend, Ind., in a Chevy powered Healy.

It was like thundersville when Al Carter's black A/Roadster tangled with Pete Van Kirk's 'T'. Al lost the race.



Two huge hunks of Pontiac and churning dual slicks denote Mr. Ed Hill of Wichita Falls, Tex. Car tore holes in the starting line.

compression almost got them into trouble with officials. On the second day of running, they were noted to be running a water injection unit. The officials asked them to remove it, which they did. As most experienced engine builders, they didn't expect great horsepower gains from the water, they just wanted to prolong the life of the high c.r. engine. At any rate, they pulled the water injector and proceeded to go even faster.

This single engine star from out of the South never once ran an elapsed time above 9 seconds. In fact, it ran several times in the 8.44 second area but these were discounted because the front wheels jumped the initial timing lights. Ollie Riley of the Chrondek Company and several of the NHRA officials teamed to get the most accurate e.t. possible on the Georgia car and came up with an official low e.t. of the meet of 8.68 seconds. From where we sat, it seems sure that Robinson can roll through the lights and still come on strong enough to get consistent 8.50 e.t.'s.

Monday was scheduled to see the eliminators crowned in Top, Middle, Little, Street, and Stock divisions. Official record runs were also open and the weather was perfect. Actually, the weather had cooperated very nicely. Dr. Nathan Ostich of Bonneville fame was on hand as official medical expert but spent most of his time on sunburns. He did treat a broken leg during the course of the meet, but that was the extent of the medical problems.

Before we forget it, we must mention the number of magnesium alloy wheels prevalent at the Nationals. Nearly every car in competition had either Halibrand or American units and the droves of Corvettes in the sports car classes were considered naked without the items. Also in prominence were drag chutes. Nearly all kinds were on hand, but the majority were of either the triangle or round ring-slot type. Jim Deist was on hand to help fellows with drag chutes.

It just so happens that the Labor Day weekend is a big one in the Midwest area as far as important sprint car races go and Indianapolis seemed alive with owners and drivers on USAC's Championship Trail. We looked up one day to see Roger Ward and Parnelli Jones at the strip and they were both having a ball. Roger was overheard to express how much he would like to have one of the hauling rails. Seems everyone is fascinated by drag racing these days.

Of course, everyone had by now pegged Robinson as the eventual Top Eliminator, but the big question was who would reign in the other classes. The competition was keen, but in Middle Eliminator it was Walt Knoch's A/Roadster. Little Eliminator went in an upset victory to Willis Ragsdale's Chevy-powered 'T' B/Roadster. Junior Garrison of Murray, Kentucky, picked up the Street Eliminator crown.

Wild cheering returned when the fifty fastest stockers came up to the line for Stock Eliminator. On one of the runs Proffitt had trouble shifting and the big bad Pontiac was out of contention. Surviving the quick elimination was Al Eckstrand in a 1961 Dodge entered by the Ramchargers club of Wayne, Michigan. Al was doing all right until he had a shifting problem against Don Nicholson. It seems the Dodge was using a three-speed transmission which is considered somewhat weak. On the run against Nicholson's Chevy Al had trouble with the pin-drive syncros and lost a heartbreaker. *But*, after a complete tear-down the Nicholson car was disqualified on several technical points. It's either factory stock, or it isn't stock. Period. Minor points, you might say, but nevertheless those are the rules and they must apply to everyone.

By midafternoon the final round for Top Eliminator had arrived and it just happened to match Robinson and Bob Carroll's "Fiat Bug," an A/Competition Coupe from Marietta, Pa. Robinson didn't get on the throttle too hard, but he still arrived at the finish line first with an elapsed time of 8.92 seconds and a top speed of 169.49 mph. Pete had actually considered not even running for the final eliminator awards because he had discovered an engine hurt on Sunday. Persuasive arguments by well-wishers convinced him to stay on, so he picked up some parts for the little Chevrolet and worked Sunday night getting ready.

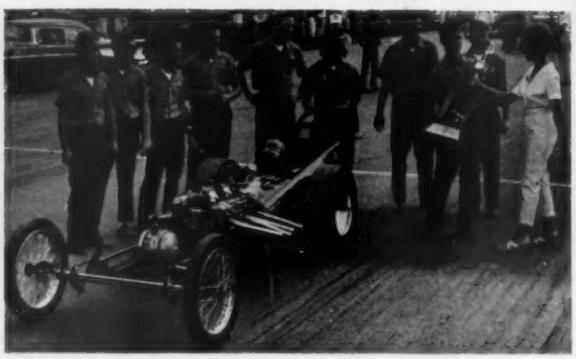
Pete Robinson had lived the final chapter of the Cinderella story of the 1961 National Championships. And what a story it was. An unknown engineer from the deep South had successfully invaded the domain of the giants and conquered all who dared approach. The National Championships, seventh edition, were over, but for Sneaky Pete the story was just beginning.

LE ROI SMITH

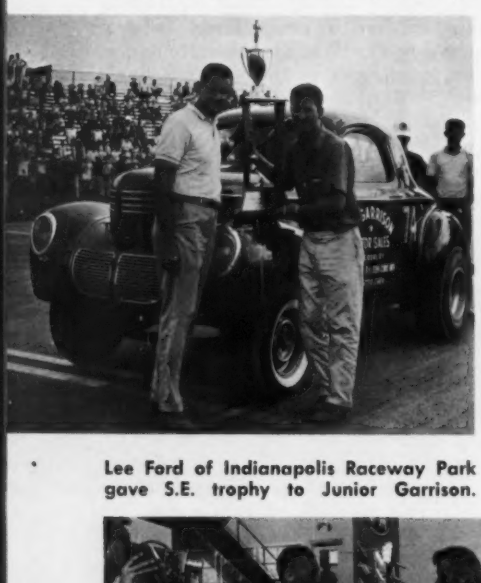
Sam Parriott, the Mayor of the City of Industry, Calif., won the Champion Spark Plug Co. trophy in A/Modified Spts.

Middle Eliminator trophy was given to Walt Knoch, left, by representative of Coca Cola Company, shown at right.

A happy Stock Points Champion Bruce Morgan receives Hurst-Campbell Pontiac Catalina sport coupe from George Hurst.

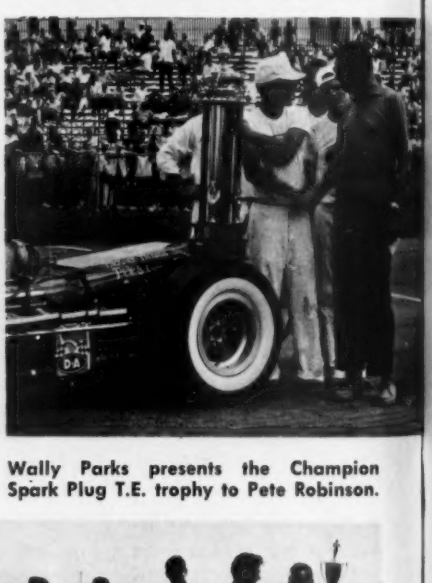


Best Appearing Car and Crew trophy was presented to Marino Monjure of Louisiana by Barbara Livingston from NHRA Hq.



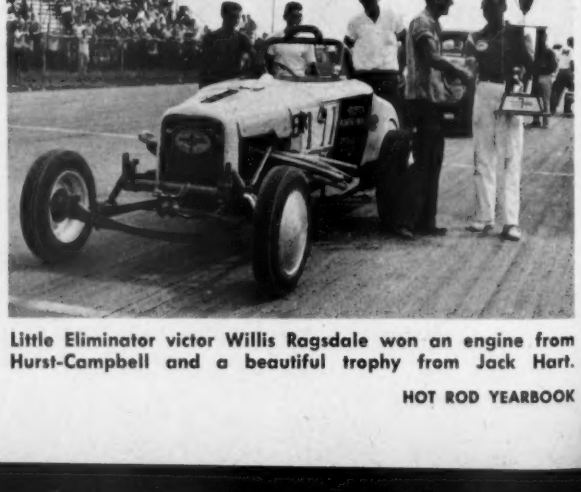
Lee Ford of Indianapolis Raceway Park gave S.E. trophy to Junior Garrison.

D-A Lubricants crew chief ring was given by Jack Martin to proud Bill Word.



Wally Parks presents the Champion Spark Plug T.E. trophy to Pete Robinson.

All winning crew chiefs got a gold torque wrench from Sturtevant Co. rep, courtesy of New Britain Machine Company.



Little Eliminator victor Willis Ragsdale won an engine from Hurst-Campbell and a beautiful trophy from Jack Hart.

1961 NATIONAL CLASS CHAMPIONS

TOP ELIMINATOR

Class-Entry-Driver-Engine:	ET:	MPH:
AA/D–51–Pete's Eng. Co. Spl., Pete Robinson, '60 Chevy, Atlanta, Georgia	8.92	169.49

MIDDLE ELIMINATOR

A/R–284–Walt's Puffer Too, Walt Knoch, Jr., Chrysler, Dearborn, Mich.	10.45	151.77

LITTLE ELIMINATOR

B/R–191–"Raunchy," Willis Ragsdale, '61 Chevy, Pasadena, Texas	11.55	122.44

STREET ELIMINATOR

B/G–74–"Wicked Willys," Jr. Garrison, '57 Chevy, Murray, Kentucky	12.44	111.66

TOP SPEED OF MEET

AA/D–4–"Two-Thing," Dode Martin, '60 Chevy (2), Carlsbad, California		177.87

LOW ELAPSED TIME OF MEET

AA/D–51–Pete's Eng. Co. Spl., Pete Robinson, '60 Chevy, Atlanta, Georgia	8.68	

GAS COUPES/SEDANS

A/G–386–"Wil-A-Meaner," Ollie Olsen, '60 Chevy, W. Palm Beach, Florida	11.68	116.27
B/G–74–"Wicked Willys," Jr. Garrison, '57 Chevy, Murray, Kentucky	12.42	112.60
C/G–227–Larry Teter, '57 Chevy, Elgin, Illinois	13.34	112.00
D/G–153–Hrudka Bros., Joe Hrudka, '61 Chevy, Parma, Ohio	12.73	105.26
E/G–500–Hart's Automotive, Jim Koonce, '57 Chevy, Cleveland, Ohio	14.07	96.20
F/G–343–"The Mau Mau," Don Neal, '50 Merc, Hurst, Texas	13.67	99.33
G/G–573–Pete McNichol, '61 Valiant, Hazel Park, Michigan	14.11	96.98

SUPERCHARGED GAS COUPES/SEDANS

A/GS–85–"World's Wildest Willys," George Montgomery, '61 Cad, Dayton, Ohio	10.91	130.63
B/GS–398–"Chicken Coupe," Ray Moore, '59 Chevy, Cleveland, Ohio	11.76	120.32
C/GS–357–Pittman-Edwards, K. S. Pittman, '53 Olds, San Fernando, California	12.03	118.89

STREET ROADSTERS

A/SR–27–"Dubble Trubble," Milton Potter, '60 Olds, Nashville, Tennessee	11.49	132.55
B/SR–367–Armstrong and Thomas, '59 Chevy, Oklahoma City, Oklahoma	12.38	114.79
C/SR–45–Brockman's Speed Shop, Jim Lightcap, '53 GMC, Dayton, Ohio	13.63	102.04

ALTEREDS

A/A–145–Ratican-Jackson-Stearns-Reath, Ron Stearns, 59 Olds, Hollywood, California	10.45	152.28
B/A–106–Waldo Hirshfield, K. S. Pittman, '60 Chevy, Paramount, California	10.88	130.24
C/A–93–"Going Goat," Billie Rasmusson, '53 Dodge, Ft. Forth, Texas	12.18	113.49
D/A–502–Richard Moreau, '56 Chevy, Hadley, Massachusetts	12.56	109.58

ROADSTERS

A/R–284–Walt's Puffer Too, Walt Knoch Jr., Chrysler, Dearborn, Mich.	10.20	155.97
B/R–191–"Raunchy," Willis Ragsdale, '61 Chevy, Pasadena, Texas	12.02	112.21
C/R–163–"Hay-Seed Special," Asch-Thomssen, '48 Ford, Lincoln, Nebraska	13.36	102.15

COMPETITION COUPES/SEDANS

A/C–103–"Fiat Bug," Custom Auto Parts, Fred Zipp, '58 Chrysler, Marietta, Pennsylvania	10.15	150.25
B/C–207–"Smarty Pants," George Weiler, '61 Chevy, Birdsboro, Pennsylvania	11.34	123.96

MODIFIED ROADSTERS

A/MR–470–"Slo Pok," Skeans-Pickering, Dave Skeans, '58 Chevy, Vancouver, Washington	9.96	145.39
B/MR–69–Van DeVenter Special, Jesse Van DeVenter, '58 Chevy, National City, California	10.64	132.61

DRAGSTERS

AA/D–51–Pete's Eng. Co. Spl., Pete Robinson, '60 Chevy, Atlanta, Georgia	8.86	170.77
A/D–100–"Mooneyes," Dean Moon, '60 Chevy, Santa Fe Springs, California	9.56	159.29
B/D–579–Starkey-Jent Spl., Chuck Starkey, Chevy, Bay Village, Ohio	10.06	149.00
C/D–165–"I Get A Hunch," Bill Schwarz, '59 Chevy, Ann Arbor, Michigan	11.05	126.93
D/D–57–Stanley Lomelino, '46 Ford, Verden, Illinois	11.58	116.27
X/D–Mickey Thompson Enterprises, Thompson, '61 Tempest, Long Beach, California	11.25	122.61

MODIFIED SPORTS CARS

AM/SP–520–City of Industry Spl., Sam Parriott, '56 Cad, City of Industry, California	11.91	128.20
BM/SP–112–"Wizard of Oz," Jerry Osborn, '57 Chevy, South Bend, Indiana	12.09	117.49
CM SP–121–Buscham-Casazza, Gus Buscham, '60 Chevy, Iona, New Jersey	13.16	107.14

All Class-Win Times Certified Accurate by NHRA and Chrondek Timing Company Officials.

1961 TOP TEN FINAL POINTS WINNERS

Competition Number	Name	Class	Points
HOT			
B0049	Jack Chrisman	AA/D	550
B0248	Custom Auto Parts	A/C	370
B0263	Henry D. Medlin	A/SR	370
B0022	Armstrong & Thomas	B/SR	360
B0258	Norman R. Ries	B/C	310
STOCK			
B0214	Bruce Morgan	B/S	450
B0318	Lennie Kennedy	D/SA	310
B0200	Harold E. Ramsey	S/S	300
B0185	Joe Burney	F/S	290
B0284	James H. Price	S/S	270

NATIONAL RECORDS

The following records were established at NHRA record events, verified for accuracy by an NHRA certification team, and were confirmed as official at time of this book's publication.

NATIONAL HOT ROD ASSOCIATION

Car Class	Entry Name	City, State	Engine Make	Drag Strip	Date	Miles Per Hr.	Elapsed Time
			DRAGSTERS				
AA/D	Dragmaster Two-Thing	Carlsbad, Calif.	2-Chevy	Indianapolis	9-61	175.78	
AA/D	Howard Cam Spl.	Los Angeles	2-Chevy	Caddo Mills	8-61		8.78
A/D	Dick Rea	Compton, Cal.	Chry	Pomona, Cal.	2-61	167.28	
A/D	Jimmy Nix	Oklahoma City	Chevy	Fort Worth, Tex.	6-61		9.17
B/D	Dan Weis	Richmond, Va.	Chevy	Emporia, Va.	5-61	145.16	
B/D	Music-Maker	Dayton, Ohio	Chevy	W. Salem, O.	5-61	145.16	
B/D	John D. Weis	Richmond, Va.	Chevy	York U.S. 30, Pa.	6-61		9.72
C/D	Walt Markert	Amherst, Mass.	DeSoto	Atco, N.J.	7-60	138.24	
C/D	Walt Markert	Amherst, Mass.	DeSoto	E. Haddam, Conn.	7-61		10.71
D/D	Westernberger, Bill Reath	Lakewood, Calif.	Merc	Pomona, Cal.	2-61		11.28
D/D	Jim Wohlfeil	Pontiac, Mich.	Ford FH	Detroit, Mich.	7-61	122.11	
X/D	Mickey Thompson	Long Beach, Cal.	Pont Temp	Pomona, Cal.	2-61	125.96	11.27
			MODIFIED ROADSTER				
A/MR	Custom Car Supply	Amarillo, Tex.	Chry	Caddo Mills	8-61	153.06	
A/MR	Skeans-Pickering	McMinnville, Ore.	Chevy	McMinnville, Ore.	8-61		9.68
B/MR	Jess Van DeVenter	National City, Calif.	Chevy	Indianapolis	9-61	132.15	10.71
			ROADSTER				
A/R	Al Carter	Oklahoma City	Chry	Indianapolis	9-61	148.51	
A/R	Al Carter	Oklahoma City	Chry	Oklahoma City	7-61		9.98
B/R	Dave Whalen, Shaker Hgts.	Ohio	Chevy	Atco, N.J.	7-60		11.45
B/R	Nick Rini	Cleveland, Ohio	'61 Chevy	Indianapolis	9-61	123.62	
B/R	Willis Ragsdale	Pasadena, Texas	'61 Chevy	Indianapolis	9-61	123.62	
C/R	Tobler & Evans	Henderson, Nev.	Buick	Pomona, Cal.	2-61	112.24	
C/R	Ray Sowers	Columbus, Ohio	Ford FH	Indianapolis	9-61		12.40
			ALTEREDS				
A/A	Walt's Puffer	Dearborn, Mich.	Chry	Indianapolis	9-61	157.06	10.49
B/A	Hirshfield-Beaver	Paramount, Calif.	Chevy	Pomona, Cal.	8-61		10.73
B/A	Hirschfield-Beaver	Paramount, Calif.	Chevy	Indianapolis	9-61	129.68	
C/A	Roger Blood	Alexandria, Va.	'61 Chevy	Indianapolis	9-61		12.11
C/A	Colon & Kovacs Spl.	Cleveland, Ohio	'60 Chevy	Indianapolis	9-61	114.06	
D/A	Richard Moreau	Hadley, Mass.	Chevy	Indianapolis	9-61	110.49	12.49
			COMPETITION COUPES/SEDANS				
A/C	Custom Auto Parts	Marietta, Pa.	Chry	York U.S. 30, Pa.	6-61	150.25	9.85
B/C	George Weiler	Birdsboro, Pa.	Chevy	Atco, N.J.	7-61	128.02	10.60
			STREET ROADSTERS				
A/SR	Boyd-Rice	Oklahoma City	Olds	Oklahoma City	7-61		11.25
A/SR	Nicholson-Coonrod	Vancouver, Wash.	Chry	McMinnville, Ore.	8-61	134.55	
B/SR	Armstrong & Thomas	Oklahoma City	Chevy	Fort Worth, Tex.	6-61		12.07
B/SR	Armstrong & Thomas	Oklahoma City	Chevy	Caddo Mills	8-61	113.06	
C/SR	J. E. (Johnny) Hart	Portland, Ore.	Ford	McMinnville, Ore.	8-61	102.38	13.29
			GAS COUPES/SEDANS				
A/G	Reath-Grist Bros.	Wilmington, Calif.	Edsel	Madera, Cal.	4-61	118.89	
A/G	Delozier-Ramey	Oklahoma City, Okla.	Pont	Roswell, N.M.	4-61		11.93
B/G	Ed Schartman	Olmsted Falls, Ohio	Chevy	Thompson, O.	6-61		12.48
B/G	Ed Schartman	Olmsted Falls, Ohio	Chevy	Detroit, Mich.	7-61	116.27	
C/G	Frank Gladney-Herbert	Baton Rouge, La.	Chevy	Caddo Mills	8-61		12.72
C/G	Ron Hassell	Parma Heights, Ohio	Chevy	Thompson, Ohio	6-61	109.35	
D/G	Hrudka Bros.	Parma, Ohio	Chevy	Thompson, O.	6-61		13.54
D/G	August Hartkopf	Austin, Texas	Chevy	Caddo Mills	8-61	104.89	
E/G	Jim Koonce	Cleveland, Ohio	Chevy	Detroit, Mich.	7-61	100.44	13.89
F/G	Don Neal	Hurst, Texas	Merc	Caddo Mills	8-61	99.22	13.41
G/G	Peter J. McNicholl	Hazel Park, Mich.	Valiant	Indianapolis	9-61	97.08	14.32
			SUPERCHARGED GAS COUPES/SEDANS				
A/GS	George Montgomery	Dayton, Ohio	'61 Cad	Indianapolis	9-61	131.77	10.83
B/GS	Stone & Woods	Los Angeles	Olds	Pomona, Cal.	2-61	127.04	11.11
C/GS	Pittman & Edwards	San Fernando, Cal.	Olds	Pomona, Cal.	2-61		11.28
C/GS	Pittman & Edwards	San Fernando, Cal.	Olds	Indianapolis	9-61	126.22	
			SPORTS CARS				
AM/SP	Sam Parriott	City of Industry, Cal.	Cad	Pomona, Calif.	2-61	130.01	11.71
BM/SP	Jerry Osborn	Mishawaka, Ind.	Chevy	Detroit, Mich.	7-61	119.36	12.07
CM/SP	Gus Buscham	Vineland, N.J.	Chevy	Indianapolis	9-61	110.83	12.62

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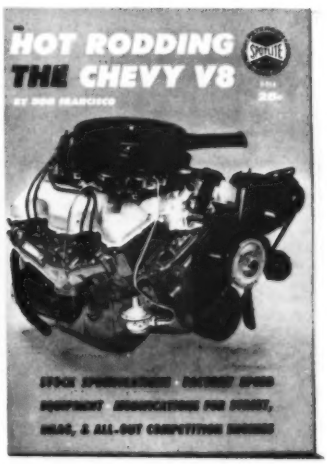

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